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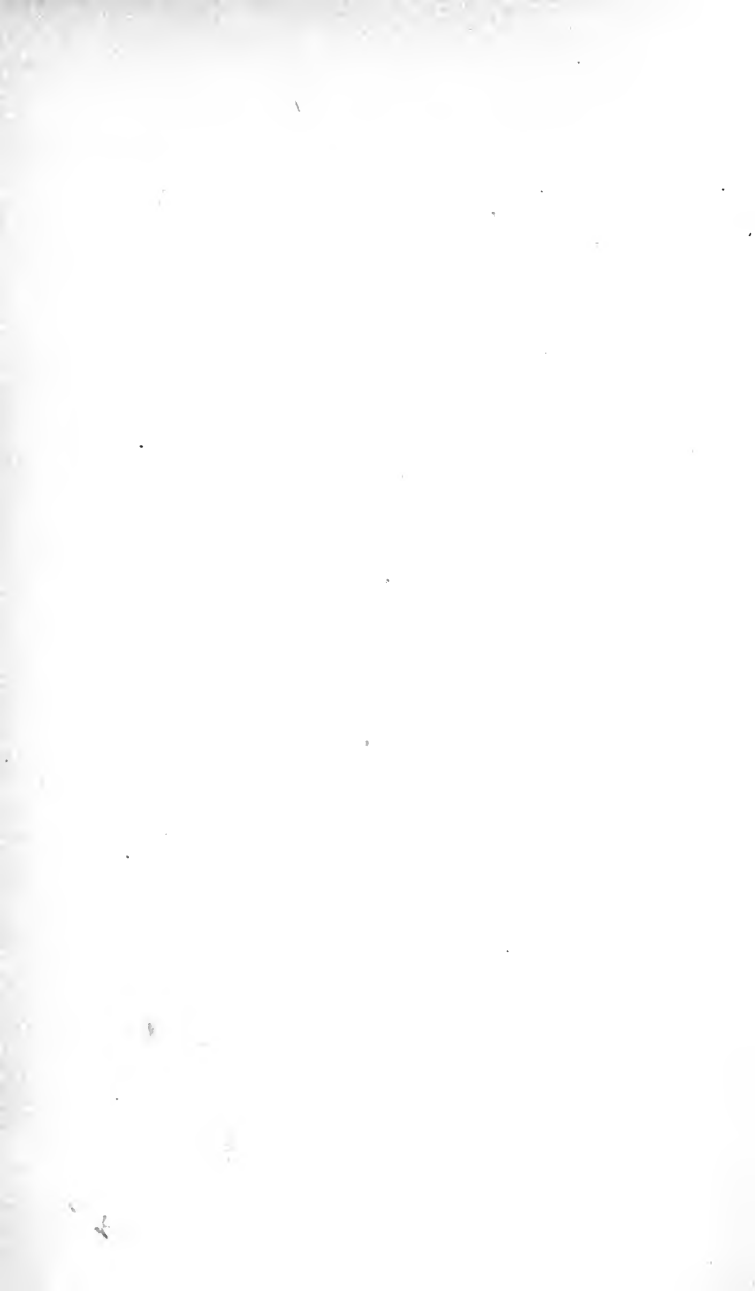
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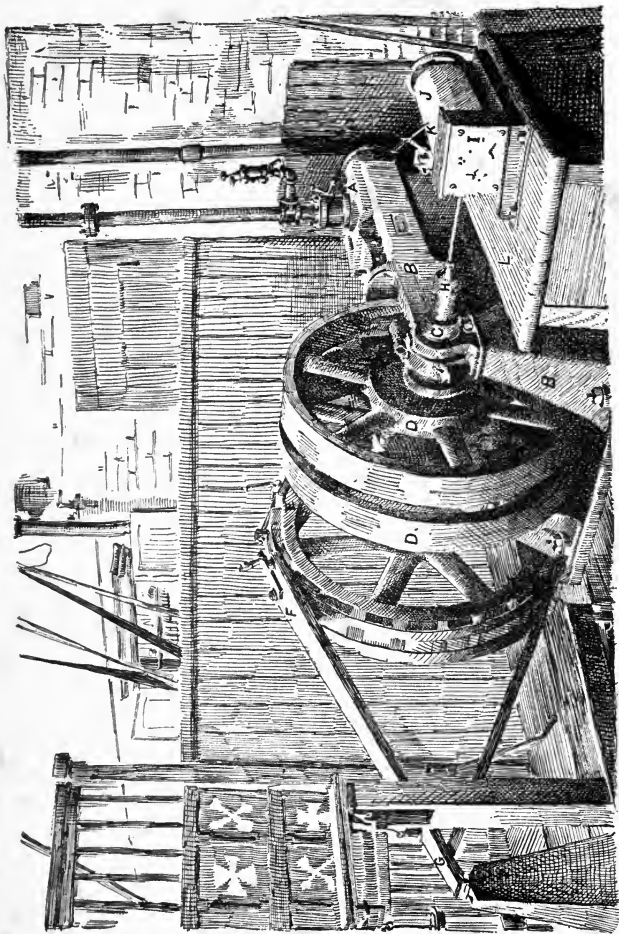
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A HANDBOOK

OF

ENGINE AND BOILER TRIALS,

AND OF THE

INDICATOR AND PRONY BRAKE.

FOR ENGINEERS AND TECHNICAL SCHOOLS.

BY

R. H. THURSTON, M.A., LL.D., DR.ENG'G;

DIRECTOR OF SIBLEY COLLEGE, CORNELL UNIVERSITY;

PAST PRESIDENT AM. SOCIETY MECHANICAL ENGINEERS;

AUTHOR OF "MATERIALS OF ENGINEERING," "FRICTION AND LOST WORK," "HISTORY OF
THE STEAM-ENGINE," "MANUAL OF STEAM-BOILERS," ETC. ETC.

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PREFACE.

THIS little treatise on methods of testing engines and boilers is an attempt to meet what has seemed to the Author a long-existing want. Hitherto, every engineer doing work of this kind has been compelled to do that work without a standard of reference, and the results of trials of engines and of boilers which have found their way into the record have been presented in such various ways as to be difficult of comparison, and such as to offer to the engineer desiring to do his work in an acceptable and permanently useful manner no generally accepted criterion. But the work of a committee of the American Society of Mechanical Engineers, of the German engineers, and of one or two individual and recognized authorities among experts, at later dates, has led to such a general concurrence among members of the profession that it is now possible to at least provisionally offer a system of testing both engines and steam-generators that may be accepted as satisfactory.

That this system will be steadily and constantly improved cannot be doubted, and the methods in vogue among the best practitioners to-day will not be precisely those in use among such experts a year hence; but the processes now adopted most generally will probably only be modified in detail, and the improvements will now mainly consist, it may be safely presumed, in the application of the most recent and accurate methods of precise measurement, as customarily applied in laboratories, to the determination of the quantities sought in such trials. The main outline of the scheme of trials to-day will be the substantial representative of similar operations later. A time has thus arrived when it is possible to put in permanent

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form, and to publish for general use, the schemes of trials which have just taken definite shape. This is what the Author has attempted to do in this treatise.

It is proposed to present here those methods of trial of heat-engines which have become standard ; to exhibit the processes of their application ; to describe the best forms of apparatus used to date in conducting them and in securing the data sought ; to illustrate their use and their various capabilities ; and, finally, to present examples of the reports made by distinguished engineers on important work of this character, and thus to give good exemplars of their form, and of the data and results deduced from them in the case of the better classes of machinery and apparatus. It is intended that the apparatus in common use shall be described, as well as the method of its application to its purposes ; its capabilities in the direct or indirect procurement of data ; the processes of computation of the latter ; the method of arrangement and tabulation of results ; and the final compilation and report on the essential quantities which are required to give basis for the determination of the economy and efficiency, physical or commercial, of the machinery employed for the development of power.

The system of boiler-trial described is that proposed by the committee above referred to, and which has since become conventionally standard throughout the United States, and largely abroad. It is more complete and more satisfactory, in the opinion of the Author, than any other yet published, and seems to have been found sufficient to meet every ordinary requirement. The special precautions advised by the several experts on the committee are also quoted, and the forms of blanks and records found best adapted for use in such work are given. Some consideration is given to the methods of determining the character and value of the steam supplied by the boiler ; and the forms of calorimeter generally used are described. It was shown by the results of a trial made by the Author for a committee of the American Institute, in 1871, that the best boilers, worked under ordinarily satisfactory conditions, give practically dry steam ; but the necessity is none the less imperative to make certain, at every important trial, that this is the case with the boilers under test. A trial made without such determination of the quality

of the steam would, to-day, be regarded by the most expert among the profession as comparatively valueless.

The text adhered to in this account of the standard boiler-trial is that of the committee, as published in the transactions of the society. References are given in all cases in which the subject is such as would justify the reader in looking up original sources of information.

The chapter on the indicator is a brief and simple account of that wonderful instrument and its capabilities, as well as a description of the usual and best ways of handling it. No attempt has been made to elaborate to any great extent the study of the diagram; but the better forms of diagram, and those which are most likely to be met with in the best, as well as those taken from some defective, engines, are illustrated. For further information upon this exhaustless subject, the reader will consult the special treatises on the steam-engine indicator, of which a number exist, each, from that of Porter, the first workmanlike presentation of the subject, up to the latest extensive work published, having its own special field and its own peculiar characteristics. All give information of real value. It has been endeavored to give some idea of the shape and of the signification of the more usual and familiar forms of "card," and to show just how they bear upon the adjustment, the proportions, and the working of the engine; while singular and rare forms, which, however, so greatly interest every engineer, are generally left to be described by the writers of the special treatises on the indicator. In this respect, the published writings of several specialists of great experience and ability in the handling of the instrument will be found peculiarly rich.

In the chapter on the measurement and computations of the indicator diagram will be found a description of the methods usually considered best and most exact, and of the processes leading to the more important of the results attainable by the use of the instrument. These are mainly well known and standard among the best practitioners; but a few are of recent application and comparatively unknown, and are here for the first time introduced into a treatise of this kind. Such are the applications of the chronograph, applied, probably for the first time, in the manner exhibited in the frontispiece, by Professor W. A.

Anthony, and used for various purposes by a few engineers when seeking to obtain precise measurements of minute and rapid variations of engine-speed. The use of the timing-fork, employed for some years past in the work of the Author, is another novel and hitherto undescribed method of securing a record of the variations of velocity of parts of the engine, and one which promises to be of great value in investigations of the character described in illustration of its use in this chapter.

The application of the Prony or dynamometric brake is still another no less important and rarely described process of securing what are now universally recognized as essential measurements in the determination of the real efficiency of the engine. The several best-known modern forms of brake are here described, and their theory and the mode of their application are given in sufficient detail, it is hoped, to enable any one to properly apply them. The transmitting dynamometer is also often now used and finds its place in the text.

Standard methods of engine-trial are as essential to the satisfactory work of the engineer as are standard boiler-trials. There exists no such precisely formulated standard for engine-as for boiler-trials; but the text includes descriptions of such methods as are, in the opinion of the best authorities, likely to give, on the whole, most satisfactory, complete, and exact results. The schemes of competitive trials as customarily conducted are presented, and examples of special methods and results are given.

The work is completed by the introduction of a series of valuable reports written by a number of the ablest members of the profession as exemplars and models of most admirable summaries of data, and of conclusions derived from their study and from the computations made therefrom. One example of each of the most important classes of engine is studied in this manner; and the series should enable any engineer unfamiliar with such work by earlier experience to secure results thoroughly satisfactory to himself and to his clients. All essential constants and tables are given in the Appendix, and should others be desired in special cases, they can be readily found—usually in the table-books which cover the desk of every engineer and every student of engineering.

The Author has sought to compile a concise, accurate, and satisfactorily complete account of the apparatus and methods of the time, as familiar to the most experienced and accurate specialists. But even as the book is going through the press, new methods are being devised and old ones improved ; while new instruments are being invented and the familiar apparatus of physical measurements are being applied to new purposes. Only the careful watch of current periodical literature will enable the engineer, young or old, to keep fully up with the progress of the age ; it is nevertheless hoped that this compilation may prove of service to many, both young and old, and may be found to include enough of the most modern and the best practice to enable its reader to attain, in his own work, satisfactory results both as to accuracy and completeness.

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ENGINE AND BOILER TRIALS.

CHAPTER I.

REASONS AND PURPOSES.

1. The Purpose of Test-trials of engines and boilers, is, commonly, the verification of the claims of the builder to complete fulfilment of his contract, and more especially as to the power and economical working of his apparatus. Whenever a motor, whether an air, a gas, or a steam or other vapor engine, is constructed for a proposing purchaser and user, the builder is expected to bind himself by a carefully drawn contract to supply apparatus capable of developing a stated amount of power and with a specified consumption of fuel, and, sometimes, of other supplies. A test-trial is demanded, when the machine is set up and in normal operation, to ascertain whether such contract and its specifications have been completely fulfilled.

In other cases, a trial is made to satisfy the proprietor that his machinery is doing good work; in still other instances, he desires to ascertain whether variations of the usual methods of operation and rules of management may be expected to give improved results; sometimes, he desires to test the skill of his men, or the character of the fuel employed. In all cases, whatever the main purpose of the operation, certain data are sought to be obtained as a basis for computation of the results needed, either to give a measure of the power and efficiency of the

machinery, or a means of comparison with other apparatus of similar character and known excellence.

A complete trial of engine and boiler involves the determination of the quantity of energy stored in potential form in the fuel; the amount liberated by combustion in available form; the proportion and the quantity taken up by the boiler; the amount stored in the steam, and in any water taken up by it, and transferred to the engine; and the distribution at the engine into useful and lost work, and wasted heat. The methods of computation of these quantities will be given presently.

The purposes and methods of such trials are thus the exact and unquestionable determination of one or several of the efficiencies of the engine—or the boiler—and these methods are usually intended to be such as will give scientifically accurate measures of the heat, the steam, the feed-water, and the energy supplied to the system; the heat, steam, and energy reaching the engine; the power developed; the distribution, usefully and wastefully, of heat, energy, or power, or of all; the power and the thermodynamic and the actual efficiency of the engine considered as a heat-engine; and also the efficiency of the engine considered as a train of mechanism, *i.e.* as a machine. It is not always essential that all these determinations shall be made, or that such as are made shall be rigidly exact. Trials are often made which give partial results, and by methods which are only approximate and sometimes but roughly approximate, if judged from the standpoint of experimental science. As in all engineering work, the ultimate gauge of expediency, as judged from a financial standpoint and with an eye to a final summation of results, determines the extent to which the engineer is justified in giving his time and incurring expense in making steam-engine and boiler trials. On extensive contracts and important and costly work, all the resources of physical science and engineering practice are applied; in minor matters, but little expense or labor is deemed justifiable.

2. Specifications of Performance, and, often, a guarantee, with forfeiture in case of non-fulfilment, should form a part of

the contract; and those assurances of efficiency should be so exact and definite that no question can arise as to their meaning and fair interpretation, when the time arrives for their verification. The customary forms of such specifications are now fairly well settled, and the usual methods of comparison and verification will be exhibited and exemplified. When no such specification exists, it is assumed that the maker is bound to do reasonably good work and to assure to the buyers reasonably good economical performance. Obvious and unquestionable delinquency, as shown by test-trial, relieves the buyer of every responsibility not specifically and unqualifiedly assumed, and throws it upon the constructor and vendor.

Engine Duty is commonly the technical measure of the efficiency of the engine as determined by the cost of the work done in fuel consumed. The "horse-power," taken in British measure as 33,000 foot-pounds per minute or 1,980,000 per hour, demands the transformation of the equivalent amount of heat into work each minute or hour; which quantity should be supplied by one-fourth of a pound of good fuel, or less than $2\frac{1}{4}$ pounds of steam, as worked in the perfect, ideal engine having an efficiency, unity. The actual consumption of energy derived from the boiler, as will be seen later, is rarely less than eight or ten times these amounts.

Pumping-engines are commonly rated by the work done by the consumption of a specified weight of fuel, as one hundred pounds. A duty of 100,000,000 foot-pounds, on this basis, would correspond to a consumption of 1.98 pounds of fuel per horse-power per hour. Mr. Sherman, assuming 90 per cent. as the efficiency of machine, including pumps as well as engine, obtains the figures in the following table (page 4).*

Mr. Emery has compared steam-engines of various kinds on the assumption that the boiler is capable of absorbing 10,000 heat-units per pound of coal consumed. This corresponds to an evaporation of 8.99 pounds of water at 80 pounds; 9.03 pounds at 60 pounds; or, 9.08 pounds at 40 pounds gauge-

* Trans. Am. Water-works Assoc., 1885.

DUTY OF PUMPING-ENGINES.

ONE MILLION GALLONS OF WATER IN 24 HOURS RAISED 200 FEET. DUTY
AND QUALITY OF FUEL PER HOUR PER HORSE-POWER.

| Duty. | Lbs. coal per million gallons. | Lbs. coal per hour per H. P. | Duty. | Lbs. coal per million gallons. | Lbs. coal per hour per H. P. | Duty. | Lbs. coal per million gallons. | Lbs. coal per hour per H. P. | Duty. | Lbs. coal per million gallons. | Lbs. coal per hour per H. P. |
|-------|-----------------------------------|---------------------------------|-------|-----------------------------------|---------------------------------|-------|-----------------------------------|---------------------------------|-------|-----------------------------------|---------------------------------|
| 30 | 5560 | 5.94 | 51 | 3271 | 3.49 | 71 | 2349 | 2.51 | 91 | 1833 | 1.96 |
| 31 | 5380 | 5.75 | 52 | 3208 | 3.43 | 72 | 2317 | 2.48 | 92 | 1813 | 1.94 |
| 32 | 5212 | 5.57 | 53 | 3147 | 3.36 | 73 | 2285 | 2.44 | 93 | 1793 | 1.92 |
| 33 | 5054 | 5.40 | 54 | 3089 | 3.30 | 74 | 2254 | 2.41 | 94 | 1774 | 1.90 |
| 34 | 4906 | 5.24 | 55 | 3033 | 3.24 | 75 | 2224 | 2.38 | 95 | 1756 | 1.88 |
| 35 | 4766 | 5.09 | 56 | 2979 | 3.18 | 76 | 2195 | 2.35 | 96 | 1738 | 1.86 |
| 36 | 4634 | 4.95 | 57 | 2926 | 3.13 | 77 | 2166 | 2.31 | 97 | 1720 | 1.84 |
| 37 | 4508 | 4.82 | 58 | 2876 | 3.07 | 78 | 2138 | 2.28 | 98 | 1702 | 1.82 |
| 38 | 4390 | 4.69 | 59 | 2827 | 3.02 | 79 | 2111 | 2.26 | 99 | 1685 | 1.80 |
| 39 | 4276 | 4.57 | 60 | 2780 | 2.97 | 80 | 2085 | 2.23 | 100 | 1668 | 1.78 |
| 40 | 4170 | 4.45 | 61 | 2734 | 2.92 | 81 | 2059 | 2.20 | 101 | 1651 | 1.76 |
| 41 | 4068 | 4.35 | 62 | 2690 | 2.87 | 82 | 2034 | 2.17 | 102 | 1635 | 1.75 |
| 42 | 3972 | 4.24 | 63 | 2648 | 2.83 | 83 | 2009 | 2.15 | 103 | 1619 | 1.73 |
| 43 | 3880 | 4.14 | 64 | 2606 | 2.78 | 84 | 1986 | 2.12 | 104 | 1604 | 1.71 |
| 44 | 3790 | 4.05 | 65 | 2566 | 2.74 | 85 | 1962 | 2.10 | 105 | 1589 | 1.70 |
| 45 | 3707 | 3.95 | 66 | 2527 | 2.70 | 86 | 1940 | 2.07 | 106 | 1573 | 1.68 |
| 46 | 3626 | 3.87 | 67 | 2490 | 2.66 | 87 | 1917 | 2.05 | 107 | 1559 | 1.67 |
| 47 | 3549 | 3.79 | 68 | 2453 | 2.62 | 88 | 1895 | 2.02 | 108 | 1544 | 1.65 |
| 48 | 3475 | 3.71 | 69 | 2417 | 2.58 | 89 | 1874 | 2.00 | 109 | 1530 | 1.63 |
| 49 | 3404 | 3.63 | 70 | 2383 | 2.55 | 90 | 1853 | 1.98 | 110 | 1516 | 1.62 |
| 50 | 3336 | 3.56 | | | | | | | | | |

pressure from a temperature of 100° F. in each case. Ten thousand heat-units per pound of coal is equivalent to one million heat-units per 100 pounds of coal, and as the duty of pumping-engines is conventionally expressed in millions of foot-pounds per 100 pounds of coal, it follows on the basis presented that *the number of foot-pounds per heat-unit represents also the number of millions of foot-pounds duty per 100 pounds of coal.** The performance of all kinds of steam-engines may be readily compared on this basis. Ten thousand heat-units per pound of coal represent an efficiency of only $(10,000 \times 100 \div 14,500 =)$ 69 per cent. of the calorific value of pure carbon and of the best anthracite; so that ordinarily more than $(100 - 69 =)$ 31 per cent. of the heat in the fuel is carried up the chimney. The mechanical equivalent of one heat-unit is 772 foot-pounds, which, on the basis above, correspond to a duty of 772 millions of foot-pounds per 100 pounds of coal. The most economical

* Centennial Report; Group XX; 1876.

steam-engines have been claimed to give as a maximum only about 130 millions, on the same basis, equivalent to an ultimate efficiency of $(130 \times 100 \div 772 =) 16.84$ per cent. of the heat in the steam, and but $(16.84 \times .69 =) 11.62$ per cent. of the calorific value of the fuel. If

D = duty in foot-pounds per 100 pounds of coal,

H = the height of lift per gauge,

T = the initial and t the final temperatures, respectively,

then

$$D = \frac{1\ 000\ 000}{T - t + .0013 H}.$$

The standard taken by British engineers in measuring duty is usually the number of pounds raised one foot either by 94 pounds of coal, or by one "hundred-weight" (112 pounds). On this basis, the duty is computed from the ascertained weight of fuel per I. H. P. per hour, thus :

D = duty in millions ;

F = weight of fuel per I. H. P. per hour ;

$$D = \frac{186.12}{F}, \text{ where the unit is 94 lbs. ;}$$

$$D = \frac{221.76}{F}, \text{ where in cwts.}$$

3. The Various Objects, in Detail, sought in test-trials are determined in part by the specific purpose of the trial ; but, in nearly every case, they require the measurement of power obtained and of cost of obtaining it, expressed either in money or in fuel consumed ; and this means the exact weighing of fuel, the measurement of water used, the determination of the quality of the fuel and of the steam made, and its quantity, and the distribution of the stored energy, by the engine, in useful and in wasteful directions. Every quantity must be exactly measured which has importance in relation to the question at issue, and the data collected must be secured in such manner that their

magnitudes may be readily introduced into the computations and may, if question arises, be readily checked and verified. The methods of determination and of record thus become matters of importance and careful study, and the application of the greatest possible ingenuity and skill are demanded in the effort to devise an acceptable and reliable system. In studying the efficiency of capital, it is first necessary to consider the elements of cost of power.

The Annual Cost of Steam-power consists:

(1) Of certain expenses, which, in any given case, are usually invariable, whether the work is done by a large engine with high ratio of expansion and small boilers, or with a smaller engine working at a low rate of expansion and with larger boilers. These are usually: rent of building or interest on cost, taxes, repairs, etc., etc., of structure and location, the engineer's salary, and sometimes all or part of the fireman's or stoker's, also sundry minor expenses or a part of each of other expenses which, as a whole, are variable. Both of the latter classes may usually be neglected in solving the problem here first considered.

(2) The interest on first cost of engine in place, the cost of repairs, and a sum which measures the depreciation in value of the machine due to its natural wear, or to its decreasing value in presence of changes that finally compel the substitution for it of an improved engine. Oil, waste, and other engineer's stores fall under this head. These items are variable with size and style of engine.

(3) The expenses of supplying the engine with steam. These are:

(a) The cost on fuel account of the steam supplied, and which includes, also, the cost of steam condensed *en route* to the engines and wasted by cylinder condensation and leakage, as well as that actually utilized. This total quantity of steam greatly exceeds that actually used in the production of power by simple transformation of heat-energy. This item varies with the efficiency of engine and size of boiler demanded.

(b) The account of interest on cost of boilers in place and of their appurtenances, rent of boiler-room, depreciation, repairs,

and insurance, which latter account is wholly chargeable to boilers. This is also variable with size of boilers.

(c) Cost of attendance in excess of the costs included in the constant quantity in item (1) and variable with size of boiler or quantity of steam demanded.

The salary of the engineer is usually not chargeable to either engine or boiler; his position is one of supervision over the whole apparatus, and a good engineer generally keeps the closest watch over the boilers. The engine can usually be trusted, much of the time, to take care of itself. With small engines, the engineer is also the fireman. With large engines, the number of regular firemen—or, at least, the number in excess of one attendant—may be taken as proportional to the quantity of steam demanded when working at ordinary power, and with very large marine engines the same remark may sometimes apply to engine-room attendance.

The object of engine and boiler trials is to determine the magnitude of those quantities in item (3) *a* which can only be determined by direct and careful measurement.

The plan of this manual comprehends a study of the accepted methods of trial of engines and boilers, of the apparatus employed, the best methods of use, and the processes of determination of exact results and reliable conclusions. The various purposes of these investigations will be described and defined, and the quantities of which measures are to be sought will be specified; the various apparatus used in the work will be enumerated, and the general plan of the trials to be studied laid down. The instruments generally employed will be described fully and their proper manipulation shown. Methods of correction, of standardization, and of elimination of errors of observation and record will be indicated, and the character of all essential operations of computation exhibited. The usually practised systems of test-trial will be described, and the work illustrated by the presentation of sample reports and of results.

4. The Maker and the Methods of the Trial are usually subjects of stipulation in the contract. In some cases the parties to the contract merely agree that any question arising shall

be left to the arbitrament of a board of three competent men ; one appointed by each, and the third by the first two appointees. In other cases, it is agreed that a known and reputable expert shall either conduct the trial or shall direct the appointments. Most frequently, when the work is very important, the contract provides for a board of three experts, who are usually named in the instrument, and also prescribes the method of conducting the trial. Those who are chosen for so important a duty should always be engineers of known ability, integrity, energy, and experience ; whose professional practice has afforded them opportunities to become familiar with the best methods and has given them skill in their employment. Standard and well known and universally accepted methods should be prescribed whenever possible.

Earlier methods and authorities should be studied by those who desire to trace the progress of the art of engine-testing ; but it will be found that very little has been done to reduce the several processes to exact form, and to a system, until very recently. A commission of eminent German engineers began such a codification of current practice, and a committee of the American Society of Mechanical Engineers has established a standard for tests of boilers. Various practitioners in this country and abroad, and the managing boards of public exhibitions and competitions, have gradually come to substantial agreement as to systems, and as to the methods which it is the purpose of this work to describe and which may be now taken as representing the most generally approved and recent practice.

5. The Character of Report demanded of the engineers conducting the trial is determined by the purpose of their work. In every case, it should be simple, easily comprehended, so far as that may be possible, by a non-expert reader ; and it should give all essential data, processes, and results, in such manner that, in case they are called in question, they may be verified completely and with certainty. The matter sought to be settled should be defined with precision and clearness, and the whole statement should be as concise and as absolutely free from irrelevant matter as possible. In the endeavor to explain

presumably obscure points, even, opportunity will be found for the exercise of a good judgment and great discretion. Good examples of the best forms of report will be presented later.

Recent standard methods have approximated very closely to the exactness and accuracy of the processes of the physicist or the chemist. In fact, the apparatus and processes of these investigators are now adopted by the practitioner, and the young engineer himself is now almost invariably trained by the physicist and chemist, in all the exact methods of the laboratory, in his preliminary professional studies in the technical school. The determination of temperatures and the ascertainment of weights are conducted by all the accepted methods of exact scientific measurement, and the analysis of fuels, of gases, and of ashes is bringing into play and application some of the most interesting and accurate operations of the chemical laboratory.

6. The Apparatus employed in test-trials of engines and boilers should be made and standardized with all the care exacted in any other department of physical research. Instruments of every class used in such investigations should be carefully selected, and from the stock of the best makers; they should be inspected, tested, and compared with standard instruments of known excellence and accuracy, and any permanent errors in their operation and method of application should be recorded with every possible care. Scales should be compared with the legal standards; tanks and other volume-measures should themselves be very nicely measured; thermometers should be calibrated; indicators should be tested, hot as well as cold; and dynamometers should be measured up with similar accuracy.

7. Methods of Application of the instruments used are commonly well determined by experience and settled by custom. Such serious variations of result may be due to error in this matter that the study of the effects of differences of practice becomes an important part of the task of the engineer-expert. A fault in location of a thermometer, or in the setting up of an indicator, may produce quite sensible, and even serious, differences in the magnitude of data so obtained.

8. The Data Needed and Computations required in the determination of the character and performance of any engine or boiler, or of both, are obtained by means of continuous or closely successive readings from all instruments employed, so taken and recorded as to give a substantially exact record of the whole period over which it has been concluded to extend the trial. These observations must be sufficiently frequent and numerous to permit the computation of very accurate averages, and, where graphical methods are, as is now common, adopted, to permit the construction of smooth and complete curves showing the whole period of the trial. Every continuous process should thus be capable of representation by a similarly continuous record, and in such manner that every computation required may be readily effected. The results of computation will then furnish accurate numerical measures of every quantity needed to determine whether the contract has been fully complied with.

9. Trials to Determine Economy and Efficiency are most common, and are generally of most importance. Few contracts for important steam apparatus are made which do not include a specification of the degree of economy demanded in the use of both steam and of fuel, and often of a system of test-trial as well. In such cases, it becomes necessary for the engineers conducting the trial to ascertain with precision the quantity of fuel, of steam, or of heat-energy supplied to the engine, the amount of energy converted into the mechanical form, the proportion of heat and of mechanical power wasted, the methods and extent of waste, in full detail, and the power applied by the machine to such useful purpose as it is designed to subserve. The measure of the benefit received by the user of the engine is the useful work performed; the measure of its cost to him is what he pays in fuel, steam, or heat-energy, and the money equivalent of this supply, and of all incidental costs, such as rent, attendance, wear and tear, insurance and taxes, and depreciation. A comparison of the mean continuous cost with the average value of power supplied for useful work exhibits the real value of the machinery to its purchaser and user.

Such a trial is only complete when it determines accurately the following :

| EXPENDITURES : Quantities and Costs. | RECEIPTS : Quantities and Costs. |
|-------------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Fuel ; or Steam ; or Heat-energy ; supplied by the user. | Useful Work ; Wasted Work and Heat : (a) Friction of Engine ; (b) Heat lost externally ; (c) Heat lost internally and rejected from the system. |

10. Steam-boiler Efficiency is not difficult of definition when the nature of the quantity to be measured is itself first understood. There are, however, as will be presently seen, several different efficiencies of the steam-boiler, as of the steam-engine ; and it is important that each be distinctly defined before a study of either, or of total efficiency, can be made. In general, it may be said that efficiency is measured by the ratio, in common or similar and definitely related terms, of a result produced to the cost of its production. As, in the study of the steam-engine, either efficiency is measured by the ratio of work done in the specified manner to the work or work-equivalent expended in doing it ; so, in the case of the steam-boiler, either efficiency is measured by the ratio of a heat-effect, or its equivalent, to the quantity of heat, actual or latent, paid for its accomplishment.

In some cases it is not practicable to thus establish a numerical value of an efficiency ; and it can only be shown that efficiency, in the sense of quantity of result compared with magnitude of means used, is increased or decreased by the operation of defined phenomena, or by conditions which can be specified. A common measure cannot always be found, or an exact law of relation established.

Increasing steam-pressure gives increasing economy up to a limit somewhere above customary pressures. The higher the pressure the greater the economic value of the steam in a steam-engine ; but on the other hand, the lower the efficiency

of the boiler; and it is perfectly possible to reach a point at which the gain on the first score is more than counterbalanced by the loss on the second. Where the object sought is simply heating-power, the advantage lies, on the whole, on the side of low pressures.

The measure of efficiency of boilers is commonly a ratio of heat applied to a defined purpose or obtained in store, in a stated form, to the total quantity of heat from which it has been saved, another part having been diverted to other purposes, and, for the use considered, wasted. Thus, a given quantity of heat being stored as potential energy of chemical action in fuel, some proportion of that energy is received at the steam-engine when that fuel is burned under a steam-boiler; the ratio of these two quantities—always a fraction and often small—is the *total* efficiency of the whole apparatus employed in the combustion of fuel, the transfer of heat-energy to the fluid in which it is stored, and its further transfer to the point at which it is usefully applied by transformation into mechanical energy and work.

The efficiency of combustion thus measures the ratio of the available heat-energy of the fuel to that set free by its union with oxygen, and is less than unity in the proportion in which the combustible portion of the fuel escapes such chemical change or is imperfectly burned, as when a part of the fuel falls into the ash-pit, is imbedded in clinker, or remains on the grate when the fire is extinguished; or as when carbon is only oxidized to carbon monoxide instead of being completely burned into dioxide. In well-managed furnaces the value of this efficiency approaches unity; it ought not to fall below 0.90, probably, in any ordinary case.

The efficiency of transfer of heat similarly measures the ratio of heat received from the furnace by the boiler to that produced by combustion. That not transferred to the boiler is either sent up the chimney, where it is, in a certain degree, useful in producing draught, or it is lost by conduction and radiation to surrounding bodies. In good examples, the value of this ratio exceeds 0.75, and it should not usually fall under

fifty or sixty per cent. Its best value depends on considerations, however, to be hereafter stated, and it is not always desirable that it should have the highest value possible, or approximate unity.

The net efficiency of boiler is the continued product of all efficiencies.

II Efficiency of Heating-Surface measures the ratio of actual amount of heat transmitted across such surface to the total quantity available for such application; in steam-boilers it is the ratio of the quantity of heat utilized in heating and vaporizing the fluid to the total which is produced by the furnace, the unutilized heat being wasted by conduction and radiation to other bodies, or sent up the chimney. An expression was found by Rankine which has been found to give very satisfactory results when properly used in application to the ordinary work of steam-boilers. This expression may be derived as below.

Let w be the weight of furnace-gases discharged per hour, $T - t$ the difference between the temperatures of gas and water on opposite sides of any part of the plate on the elementary area dS , C the specific heat of the gas, and let q be the quantity of heat passing across unity of area in unity of time for a difference in temperature $T - t$, in other words, the "rate of conduction" per unit of area per hour.

The quantity of heat transferred across the area dS is then equal to qdS , and the fall of temperature of gas must be this quantity divided by the product of the weight, w , and specific heat, C , of the gas from which the heat is derived,

$$\frac{qdS}{Cw} = -dT;$$

and the gas flows on to the next elementary area and beyond, surrendering its heat as it goes, until it finally leaves the absorbing surface and enters the chimney-flue.

If T_1 and T_2 are the initial and final temperatures of the gas, and t the temperature of the water entering the boiler, the

heat produced, Q_1 , and that wasted, Q_2 , per hour, are respectively measured by

$$Q_1 = Cw(T_1 - t) : Q_2 = Cw(T_2 - t), \text{ nearly;}$$

while the efficiency of the heating-surface is measured by the ratio of total heat to absorbed heat; or, if the feed enters at atmospheric temperature, or nearly so, by

$$\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1 - t_1}, \text{ nearly.}$$

For the every-day work of the engineer dealing with familiar forms of steam-boiler under the ordinary conditions of daily operation, it is often most satisfactory to make use of empirical expressions derived from experiments conducted with boilers of similar character and under similar conditions of use. The expression elsewhere given as constructed by Rankine for efficiency of furnace is a rationally produced algebraic statement of a law, for example, which may also be approximated with quite sufficient accuracy, by direct determination, for any specified case. Should an error be made in assigning the correct coefficient, in the former, it may produce errors of serious magnitude; if the empirical expression be applied to other than the conditions on which it was based, errors will certainly arise which may become important.

But that expression (§ 11) and other formulas which are more or less empirical may be relied upon wherever the conditions under which it is proposed to apply them are substantially the same as those under which they were constructed and those which furnished the constants which enter into them. Where these conditions are precisely the same, empirical expressions like those of Emery (p. 19) and others are often considered even more satisfactory than any rational forms yet obtained; the reason of this fact being that the actual conditions of operation are usually not precisely those assumed by the author of the formula.

The heat utilized, $Cw(T_1 - T_2)$, is also equal to that absorbed and transmitted, qdS :

$$\int qdS = Cw(T_1 - T_2) \text{ and } \frac{S}{Cw} = \int_{T_2}^{T_1} \frac{dT}{q}.$$

The value of q has been found to be well represented by equation (4) of article 95, in which $q = \frac{Q}{Ap}$, and hence $q = \frac{(T_1 - T_2)^2}{a}$; and thus

$$\frac{S}{Cw} = \int_{T_2}^{T_1} \frac{dT}{q} = a \int_{T_2}^{T_1} \frac{dT}{(T-t)^2}.$$

Assume $(T-t) = x$, then

$$\begin{aligned} \frac{S}{aCw} &= \int_{T_2}^{T_1} \frac{d(T-t)}{(T-t)^2} = \int_{T_2}^{T_1} x^{-2} dx \\ &= \left[-x^{-1} \right]_{T_2}^{T_1} = \left[\frac{1}{T-t} \right]_{T_2}^{T_1}; \end{aligned}$$

$$\therefore \frac{S}{aCw} = \frac{1}{T_2-t} - \frac{1}{T_1-t} = \frac{(T_1-t) - (T_2-t)}{(T_1-t)(T_2-t)},$$

and the efficiency becomes

$$E = \frac{T_1 - T_2}{T_1 - t} = \frac{S}{aCw} (T_2 - t).$$

Then, since

$$\frac{T_1 - t}{T_2 - t} = \frac{S(T_1 - t) + 1}{aCw} = \frac{S(T_1 - t) + aCw}{aCw},$$

and

$$\frac{(T_1 - t) - (T_2 - t)}{T_2 - t} = \frac{T_1 - T_2}{T_1 - t},$$

$$E = \frac{T_1 - T_2}{T_1 - t} = \frac{S(T_1 - t)}{S(T_1 - t) + aCw}.$$

If the total heat absorbed per hour be taken as H ,

$$H = Cw(T_1 - t); \quad T_1 - t = \frac{H}{Cw};$$

and a simplified expression,

$$E = \frac{S}{S + \frac{aC^2w^2}{H}},$$

is obtained, in which Cw may be taken as proportional to the weight of air supplied or of fuel burned, and H as proportional to the same quantity. Thus if F is the weight of fuel burned in the given time, on unity of grate-area, the efficiency may be expressed as

$$E = \frac{BS}{S + AF} = \frac{B}{1 + AR},$$

which is the formula sought. A and B are constants to be obtained by experiment for the special type of boiler to be considered.

When S and F represent respectively the number of square feet of heating-surface per square foot of grate in any boiler, and the number of pounds of fuel burned as the square foot of grate per hour, and $R = \frac{F}{S}$, the values of A and B , as given by Rankine,* are as follows:

* Steam-engine, p. 294.

| BOILER TYPE. | | | | | A. | B. |
|--------------|------------------|----------------------|--|--|-----|------|
| Class 1. | Best convection, | chimney draught..... | | | 0.5 | 1.00 |
| " 2. | Ordinary | " " " | | | 0.5 | 0.90 |
| " 3. | Best | " forced " | | | 0.3 | 1.00 |
| " 4. | Ordinary | " " " | | | 0.3 | 0.95 |

These constants are derived from experience with good fast-burning bituminous coals; for anthracites of good quality the Author has usually found the following values more in accordance with good practice :

| BOILER TYPE. | | | | | A. | B. |
|--------------|--|--|--|--|-----|------|
| Class 1..... | | | | | 0.5 | 0.90 |
| " 2..... | | | | | 0.5 | 0.80 |
| " 3..... | | | | | 0.3 | 0.90 |
| " 4..... | | | | | 0.3 | 0.85 |

When feed-water heaters are used, or superheaters are employed, their surface should be included in the area *S*. The formula assumes no loss by excess of air-supply. Where such excess is noted or anticipated, it may be allowed for by increasing the value of *A* in proportion to the square of the total quantity of air supplied. The following table presents values of efficiency for a wide range of practice :

EFFICIENCY OF BOILERS.

| R. | BITUMINOUS COAL. Class of Boiler. | | | | ANTHRACITE COAL. Class of Boiler. | | | |
|-------|--------------------------------------|------|------|------|--------------------------------------|------|------|------|
| | I. | II. | III. | IV. | I. | II. | III. | IV. |
| 10 | 0.16 | 0.15 | 0.25 | 0.22 | 0.14 | 0.14 | 0.23 | 0.20 |
| 4 | 0.33 | 0.31 | 0.45 | 0.43 | 0.30 | 0.28 | 0.40 | 0.39 |
| 2 | 0.50 | 0.46 | 0.62 | 0.59 | 0.45 | 0.50 | 0.56 | 0.53 |
| 1 | 0.66 | 0.61 | 0.77 | 0.73 | 0.60 | 0.55 | 0.70 | 0.66 |
| 0.80 | 0.71 | 0.65 | 0.81 | 0.77 | 0.64 | 0.59 | 0.73 | 0.69 |
| 0.67 | 0.75 | 0.69 | 0.83 | 0.79 | 0.67 | 0.63 | 0.75 | 0.72 |
| 0.50 | 0.80 | 0.73 | 0.87 | 0.83 | 0.72 | 0.65 | 0.78 | 0.75 |
| 0.40 | 0.83 | 0.76 | 0.89 | 0.85 | 0.75 | 0.68 | 0.80 | 0.77 |
| 0.333 | 0.86 | 0.80 | 0.90 | 0.86 | 0.77 | 0.72 | 0.81 | 0.78 |
| 0.167 | 0.93 | 0.85 | 0.95 | 0.90 | 0.84 | 0.77 | 0.86 | 0.81 |
| 0.111 | 0.95 | 0.87 | 0.97 | 0.92 | 0.86 | 0.78 | 0.88 | 0.83 |

These values have been found to agree well with practice up to rates of combustion exceeding 50 or 60 pounds per

square foot of grate-surface per hour, beyond which point the efficiency falls off. But agreement can only be expected where the combustion and air-supply are in accordance with the assumptions on which the formula is based.

The problem of the designer of steam-boilers often takes the form: Required to determine the area of heating-surface needed to secure a stated efficiency. In this case the formula above given must be transformed thus:

$$E = \frac{B}{1 + AR} = \frac{B}{1 + \frac{F}{S}};$$

$$S = \frac{F}{\frac{B}{E} - 1}; \dots \dots \dots (13)$$

$$R = \frac{S}{F} = \frac{1}{\frac{B}{E} - 1}; \dots \dots \dots (14)$$

from which expressions, the efficiency aimed at being given, the ratio of heating to grate-surface and the extent of heating-surface may be computed. As will be seen later, the question to what extent efficiency may be economically carried by extending heating-surface is one of the problems arising in designing boilers.

Mr. Emery finds that the results of extensive series of experiments on good forms of boiler may be represented by the empirical expression, for vertical tubular boilers,

$$E = \frac{46.045}{c + 3.016} - 1.067;$$

in which E is the ratio of weight of water evaporated into dry steam, "from and at the boiling-point," under atmospheric pressure, to weight of fuel used; and c is the weight of combustible consumed per square foot of heating surface per hour, in pounds.

For horizontal boilers,

$$E = \frac{27.287}{c + 2.04} + 0.824.$$

The evaporation is here the maximum practicable with good anthracite coal.

The maximum efficiency is given as

$$E_e = \frac{e}{E};$$

in which e is the observed evaporation, reduced to the standard basis. These relations are shown in the table on p. 20. For badly designed or mismanaged boilers deduction must be made from the efficiencies and evaporations here given, to the extent of ten per cent or more, according to magnitude of the defect.

In land boilers, it is customary to keep the rate of combustion per square foot of grate down to about eight pounds per hour, although it frequently rises to 10 and 12. In marine boilers this rate is increased to 12 and 16 pounds per square foot of grate per hour when anthracite coal is burned with natural draught, and to 20 pounds and upward per hour for bituminous coal. In a locomotive, however, with forced draught, 75 and 100 pounds of coal are burned per square foot of grate. Apparently no losses result from such variations in the size of the grate, and, in fact, it appears indisputably that with a reduced grate and forced draught the air-dilution is reduced and the evaporation therefore somewhat increased. It is reasonable to conclude, therefore, that when care is taken to insure perfect combustion by ordinary tests, the relative area of the grate upon which the coal is consumed does not affect the result, and economy depends under proper conditions upon the rate of combustion per unit of heating surface as stated.

12. Effective Development, Transfer, and Storage of Heat, in the best possible combination, is evidently what is demanded in the operation of the steam-boiler.

In securing complete combustion, an ample supply of air

PERFORMANCE OF BOILERS.*—EMERY.

| 1 | 2 | 3 | 4 | 5 | 6 |
|----------------------------------------------------------|--------------------------------------------------------------------|-------------------------------------|----------------------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|----------------------------------|
| <i>c</i> | <i>E</i> | <i>cE</i> | <i>E</i> + 15 | $34.52 + 1.2 + E$ | $34.52 + cE$ |
| Combustible consumed per square foot of heating surface. | Water evaporated at atmospheric pressure from temperature of 212°. | | Ultimate efficiency. | Coal (with 1-6 refuse) per horse-power per hour. | Heating surface per horse-power. |
| | Per pound of combustible. | Per square foot of heating surface. | | On basis that one horse-power requires 30 pounds of water per hour, evaporated at 70 pounds pressure from temperature of 100°, or 34.52 pounds at atmospheric pressure from temperature of 212°. | |
| Pounds. | Pounds. | Pounds. | | Pounds. | Square Feet. |
| Minimum. | 14.20 | | 0.95 | .. | |
| 0.1 | 13.71 | 1.37 | 0.91 | 3.02 | 25.18 |
| 0.2 | 13.25 | 2.65 | 0.88 | 3.13 | 13.03 |
| 0.3 | 12.82 | 3.85 | 0.85 | 3.23 | 8.98 |
| 0.4 | 12.41 | 4.96 | 0.83 | 3.34 | 6.95 |
| 0.5 | 12.03 | 6.02 | 0.80 | 3.44 | 5.74 |
| 0.6 | 11.68 | 7.01 | 0.78 | 3.55 | 4.92 |
| 0.7 | 11.32 | 7.92 | 0.75 | 3.66 | 4.36 |
| 0.8 | 11.00 | 8.80 | 0.73 | 3.77 | 3.92 |
| 0.9 | 10.69 | 9.62 | 0.71 | 3.87 | 3.59 |
| 1.0 | 10.39 | 10.39 | 0.69 | 3.99 | 3.32 |
| 1.5 | 9.13 | 13.70 | 0.61 | 4.54 | 2.52 |
| 2.0 | 8.11 | 16.22 | 0.54 | 5.11 | 2.13 |
| 2.5 | 7.28 | 18.20 | 0.49 | 5.69 | 1.90 |
| 3.0 | 6.57 | 19.71 | 0.44 | 6.30 | 1.75 |
| 3.5 | 6.00 | 21.00 | 0.40 | 6.90 | 1.64 |
| 4.0 | 5.50 | 22.00 | 0.37 | 7.53 | 1.57 |
| 4.5 | 5.06 | 22.77 | 0.34 | 8.19 | 1.52 |
| 5.0 | 4.68 | 23.40 | 0.21 | 8.85 | 1.48 |

and its thorough intermixture with the combustible elements of the fuel are essential; for the second, high temperature of furnace, it is necessary that the air-supply shall not be in excess of that absolutely needed to give complete combustion. The efficiency of a furnace burning fuel completely is measured by

$$E = \frac{T - T'}{T - t},$$

* *Sci. Am. Supplement*, No. 687, p. 10 972.

in which E represents the ratio of heat utilized to the whole calorific value of the fuel; T is the furnace-temperature; T' , the temperature of the chimney; and t , that of the external air. Hence the higher the furnace temperature and the lower that of the chimney, the greater the proportion of available heat.

It is further evident that, however perfect the combustion, no heat can be utilized if either the temperature of chimney approximates to that of the furnace, or if the temperature of the furnace is reduced by dilution approximately to that of the chimney. Concentration of heat in the furnace is secured, in some cases, by special expedients; as by heating the entering air, or, as in the Siemens gas-furnace, heating both the combustible gases and the supporter of combustion. Detached fire-brick furnaces have an advantage over the "fireboxes" of steam-boilers in their higher temperature; surrounding the fire with non-conducting and highly-heated surfaces is an effective method of securing more perfect combustion and high furnace-temperature.

In arranging heating-surface, the effort should be to impede the draught as little as possible, and so to place them that the circulation of water within the boiler should be free and rapid at every part reached by the hot gases.

The directions of circulation of water on the one side and of gas on the other side the sheet should, whenever possible, be opposite. The cold water should enter where the cooled gases leave, and the steam should be taken off farthest from that point. The temperature of chimney-gases has thus been reduced by actual experiment to less than 300° Fahr., and an efficiency equal to 0.75 to 0.80 the theoretical is attainable.

The extent of heating-surface simply, in all of the best forms of boiler, determines the efficiency, and the disposition of that surface in such boilers seldom affects it to any great extent. The area of heating-surface may also be varied within wide limits without greatly modifying efficiency. A ratio of 25 to 1 in flue and 30 to 1 in tubular boilers represents the relative area of heating and grate surfaces in the practice of the best-known builders. This proportion may be often settled by exact calculation.

The material of the boiler, as will be shown later, should be tough and ductile iron, or, better, a soft steel containing very little carbon and thoroughly homogeneous.

The factor of safety is very often too low. The boiler should be built strong enough to bear a pressure at least six times the proposed working-pressure; as the boiler grows weak with age, it should be occasionally tested to a pressure far above the working-pressure, which latter should be reduced gradually to keep within the bounds of safety. The factor of safety is seldom more than four in new boilers; and even this is reduced practically by the operation of the inspection laws.

Effective development of heat is secured primarily by the selection of good fuel, by which is usually meant fuel which consists, to the greatest possible extent, of available combustible material; but for the purposes of the engineer who designs the boiler, or of the owner for whom it is to be constructed, the real criterion of quality is the quantity of heat which the combustible, as burned in the furnace, will yield for any given sum of money expended in obtaining that heat. The cost of a fuel to the consumer consists, not simply of money paid for it to the dealer who supplies it, but also of cost of transportation and of placing in the grate, of removal of ash, of incidental expenses inseparable from its use, such as injury to boilers and other property, increased risks, and other such expenses, many if not most of which are very difficult of determination with any satisfactory decree of accuracy. Other things being equal, that fuel which gives the greatest quantity of available heat for the total money expenditure is that which permits most effective development in the sense here taken. Effective heat-development from any selected fuel is secured, as already stated, by its complete combustion in such manner as to give the highest possible temperature.

Effective transfer of heat is secured by such a form of steam-generator, and such extent and disposition of "heating-surfaces," as will most completely utilize the heat developed in the furnace and flues by causing it to flow, with the least possible loss, into the water and steam contained within the boiler; and this is effected by proper arrangement of surfaces absorb-

ing heat from the gases and yielding it to the liquid as already generally described.

Effective storage of heat can be secured by providing large volumes of water and of steam, within which the heat transferred from the furnace and flues can be stored, and by carefully protecting the whole heated system from waste by conduction or radiation to adjacent bodies. Where the demand is steady, and the supply from the fuel fairly steady also, the amount stored need not be great, as the use of the reservoir is simply that of a regulator between furnace and engine or other apparatus receiving it; but where either supply or demand is variable, considerable storage capacity may be needed.

13. Efficient Utilization of Heat is as essential to the satisfactory working of any system of generation and application of heat as is efficient production, transfer, and storage. The mode of attaining maximum efficiency depends upon the nature of the demand and the method of expenditure; and the consideration of this subject in detail would be here out of place. In general it may be said that where the heat and steam are required for the impulsion of an engine, the higher the safe pressure and the practically attainable temperature at which the supply is effected, the more efficient the utilization of the heat. These limits of temperature and pressure are the higher as the actual working conditions are made the more closely to approximate to the ideal conditions prescribed by pure science.

Where heating simply, without transformation into work, is intended, the principal and only very important requisite, usually, is to provide such thorough protection for the system of transfer and use, that no wastes of importance can take place by radiation or conduction. The character of the steam made, as to humidity, is in this case comparatively unimportant; but in the preceding case it will be found essential that it should be always dry, and it is often much the better for being superheated considerably above the boiling-point due to its pressure.

The actual standing of the best steam-engine of the present time, as an efficient heat-engine, is really very high. The sources of loss are principally quite apart from the principles of design and construction, and even from the operation of the

machine; and it may be readily shown that, to secure any really important advance toward theoretical efficiency, a radical change of our methods must be adopted, and probably that we must throw aside the heat-engine in all its forms, and substitute for it some other apparatus by which we may utilize some mode of motion and of natural energy other than heat.

The very best classes of modern steam-engines very seldom consume less than two pounds (0.9 kilog.) of coal per horse-power per hour, and it is a good engine that works regularly on three pounds (1.37 kilog.).

The first-class steam-engine, therefore, yields less than 10 per cent of the work stored up in good fuel, and the average engine probably utilizes less than 5 per cent. A part of this loss is unavoidable, being due to natural conditions beyond the control of human power, while another portion is, to a considerable extent, controllable by the engineer or by the engine-driver. Scientific research has shown that the proportion of heat stored up in any fluid, which may be utilized by perfect mechanism, must be represented by a fraction, the numerator of which is the range of temperature of the fluid while doing useful work, and the denominator of which is the temperature of the fluid when entering the machine, measured from the "absolute zero."

Thus, steam, at a temperature of 320° Fahr., being taken into a perfect steam-engine, and doing work there until it is thrown into the condenser at 100° Fahr., would yield $\frac{320 - 100}{320 + 461} = 0.28 +$, or rather more than one fourth of the work which it should have received from each pound of fuel. The proportion of work that a non-condensing but otherwise perfect engine, using steam of 75 pounds (5 atmos.) pressure, could utilize would be $\frac{320 - 212}{320 + 461} = 0.14 = \frac{1}{7}$; and, while the perfect condensing engine would consume two thirds of a pound (0.3 kilog.) of good coal per hour, the perfect non-condensing engine would use $1\frac{1}{3}$ pounds (0.6 kilog.) per hour for each horse-power developed, the steam being taken into the engine and exhausted at the temperatures assumed above.

Also, were it possible to work steam down to the absolute zero of temperature, the perfect engine would require but 0.19 pound (0.09 kilog.) of similar fuel.

It may therefore be stated, with a close approximation to exactness, that of all the heat derived from the fuel about seven tenths is lost through the existence of natural conditions over which man can probably never expect to obtain control, two tenths are lost through imperfections in our apparatus, and only one tenth is utilized in even good engines. Boiler and engine are intended to be included when writing of the steam-engine above. In this combination a waste of probably two tenths at least of the heat derived from the fuel takes place in the boiler and steam-pipes, on the average, in the best of practice, and we are therefore only able to anticipate a possible saving of $0.2 \times 0.75 = 0.15$, about one sixth of the fuel now expended in our best class of engines, by improvements in the machine itself. The best steam-engine, apart from its boiler, therefore, has 0.85, about five sixths, of the efficiency of a perfect engine, and the remaining sixth is lost through waste of heat by radiation and conduction externally, by condensation within the cylinder, and by friction and other useless work done within itself. It is to improvement in these points that inventors must turn their attention if they would improve upon the best modern practice by changes in construction.

To attain further economy, after having perfected the machine in these particulars, they must contrive to use a fluid which they may work through a wider range of temperature, as has been attempted in air-engines by raising the upper limit of temperature, and in binary vapor engines by reaching toward a lower limit, or by working a fluid from a higher temperature than is now done down to the lowest possible temperature. The upper limit is fixed by the heat-resisting power of our materials of construction, and the lower by the mean temperature of objects on the surface of earth, being much lower at some seasons than at others. In the boiler the endeavor must be made to take up all the heat of combustion, sending the gases into the chimney at as low a temperature as possible, and securing in the furnace perfect combustion without excess of air-supply.

Good furnace management, to secure maximum heat-supply from the unit weight of fuel, is evidently as essential to economy and efficiency of steam production as choice of proper fuels.

In the management of the furnace the effort should be made to secure the best conditions for economy, and as nearly as possible perfect uniformity of those conditions. The fuel should be spread over the grate very evenly, and the tendency to burn irregularly, and especially into holes or thin spots, should be met by skilful "firing," or "stoking," as it is also termed, at such intervals as may by experience be found best. The smaller the coal, where anthracite is used, the thinner should be the fire; the stronger the draught, the thicker the bed of fuel, of whatever kind. With too thin a fire, the danger arises of excess of air-supply; with too heavy a fire, carbon monoxide (carbonic oxide) may be produced. In the former case combustion will be complete, but the heat generated will be distributed throughout the diluting excess of air, and thus rendered less available, and the efficiency of the furnace will be correspondingly reduced; while in the latter case a loss arises from incomplete combustion, and waste takes place by the passage of combustible gas up the chimney. The second is the less common cause of loss of the two, but both are liable to arise in almost any boiler, and we may even have both losses exhibited in the same boiler and at the same time. Successful working demands a very perfect mixture of the combustible with the supporter of combustion, and should this not be secured, serious waste will take place.

The appearance of smoke at the chimney-top is not always indicative of serious loss, nor is its non-appearance always proof of complete combustion. With soft coals and other fuels containing the hydrocarbons some smoke usually accompanies the best practically attainable conditions; anthracites, charcoal, and coke never produce true smoke. Attempts to improve the efficiency of a heat-generating apparatus by "burning the smoke" usually fail by introducing such an excess of air as to cause a loss exceeding that before experienced from the forma-

tion of smoke. Thorough intermixture of a minimum air-supply with the gases distilled from the fuel is the only means of attaining high efficiency.

In firing, or stoking, especial care should be taken to see that the sides and corners of the grate are properly attended to. Regulation of the fire is best secured by the careful adjustment of the damper. The manipulation of the furnace doors for this purpose is likely to cause waste. Liquid fuels are especially liable to waste by excessive air-supply, and gaseous fuel exhibits a peculiar liability to the opposite method of loss; both should be, if possible, even more carefully handled than any solid fuels.

14. Trials to Ascertain Power or Maximum Capacity to do useful work are often, perhaps usually, made under the same contract as are those to determine efficiency of engine. Steam machinery is commonly guaranteed, both as to economy and capacity. Such trials are sometimes made at the same time with efficiency-trials; but the maximum power of engines and boilers is seldom, if ever, that at which best economy is obtainable. A single trial is made when the power guaranteed is that of normal working and that for which the guarantee of economy is made by the contract. A trial for capacity simply, is one in which the power only need be measured, and its cost, unless specifically demanded, is not determined. The methods employed, so far as they go, are the same as in the preceding and more complete kind of trial.

The actual power of steam and of boilers evidently depends upon the efficiency of the method of application, and on the apparatus employed. The quantity of heat-energy supplied to the engine and yielded by the generator has been seen to be easily calculable by simply multiplying the quantity of heat given to the steam, by the fuel, by the mechanical equivalent of heat. The amount available as energy may be the total quantity so supplied, as when the steam is condensed in heating buildings or otherwise, and is returned as feed-water to the boilers; or it may be any less amount, according as the method of utilization is more or less effective. The tables given in the

Appendix furnish the data for calculation in any case in which the efficiency of transfer and of transformation is known. Where no constant value can be assumed for the efficiency of the system employed, it is sometimes, nevertheless, found to be important to establish a standard conventionally. Thus, in the calculation of available stored energy, as given in the Appendix, it was assumed that the steam would be expanded to atmospheric pressure. Similarly, convention has established the unit horse-power of steam-boilers, in order to afford a standard of comparison in test-trials, and to give a means of rating boilers by the designer, the builder, or the purchaser and user.

The operation of boilers occurs under a wide range of actual conditions—the steam-pressure, the temperature of feed-water, the rate of combustion and of evaporation, and, in fact, every other variable condition, differing in any two trials to such an extent that direct comparison of the totals obtained, as a matter of information regarding the relative value of the boilers, or of the fuel used, becomes out of the question. It has hence gradually come to be the custom to reduce all results to the common standard of weight of water evaporated by the unit-weight of fuel, the evaporation being considered to have taken place at mean atmospheric pressure, and at the temperature due that pressure, the feed-water being also assumed to have been supplied at the same temperature. This, in technical language, is said to be the “equivalent evaporation from and at the boiling-point” (212° F., 100° Cent.). This standard has now become generally incorporated into the science and the practice of steam-engineering. The “Unit of Evaporation” is one pound of water at the boiling-point, evaporated into steam of the same temperature. This is equivalent to the utilization of 966, nearly, British thermal units per pound of water so evaporated. The economy of the boiler may thus be expressed by the number of units of evaporation obtained per pound of combustible.

Newcomen used steam of barely more than atmospheric pressure, and raised 105,000 pounds of water one foot high, with a pound of coal consumed. Smeaton raised the steam-pressure to eight pounds, and increased the duty to 120,000. Watt

started with a duty of double that of Newcomen, and raised it 320,000 foot-pounds per pound of coal, with steam at ten pounds. To-day, Cornish engines of the same general plan as those of Watt, but worked with forty to sixty pounds pressure, expanding three to six times, bring up the duty to 600,000 foot-pounds; while more modern compound engines have boilers carrying 150 pounds (ten atmospheres) above the normal air pressure, and the duty has been since raised to above 1,200,000 foot-pounds per pound of fuel used.

15. The Quantities Measured and Results sought to be secured are thus, in detail, as follows for any complete trial :

When the trial includes, as is most frequently the case, a trial of the boiler, the combined efficiency of boiler and engine being the final determination, arrangements must be made in advance to ascertain exactly the weights of fuel, gross and net, coal and ash for example; the weight of water supplied as "feed;" the weights, temperatures, and pressures of dry steam, and weight of entrained water; the temperatures of furnace, flues, and chimney; of superheating steam, if it be so heated; the power of the engine, gross and net; the friction of engine; the wastes by cylinder-condensation and otherwise; the steam-pressure in boiler and steam-chest; and the continually varying pressures in the working cylinder throughout the whole cycle, revolution by revolution, of the engine. Each of these quantities is measured at specified intervals, and a comparison of mean values of power usefully applied, and of expenditures made to produce it, gives the measure of the economy attained.

At the same time that the gross power developed by the action of the steam in the cylinder, the indicated power, is measured, the diagrams taken furnish the means of ascertaining precisely how the pressures and volumes of the steam simultaneously vary within the engine, and thus give a clue to, and usually a fairly exact determination of, the setting and motion of the valves and the extent to which the distribution of steam is such as will best conduce to economical working. These diagrams also enable the engineer to compute with considerable accuracy the volumes and weights of the steam at any

point, and at every point, in the stroke. A comparison of the quantities so calculated with the actual measures obtained at the boiler, or before the steam enters the cylinder, gives the measure of the quantity condensed in the cylinder, as the piston moves forward, and of the later re-evaporation. The cylinder-wastes are thus also determinable with a fair degree of accuracy. These "cards" also exhibit the amount of back-pressure, and this measures the resistances in the exhaust passages and at the condenser, if there be one, and thus afford a means of criticism of the design and construction of the engine in this respect. Similarly, the difference between the steam-pressures in the cylinder and in the steam-chest and the exhaust-chest, is a measure of the losses in the steam passages.

The usual rates of evaporation and the effect of varying the proportions of tubes has been well determined by the experiments of Isherwood and others.

The proportions of flues and tubes vary somewhat in practice; but it will be found seldom advisable to make tubes more than 50 or 60 diameters in length. Where the heating-surface consists principally of tubes, the efficiency will be found to vary with their length nearly as follows:

| | | | | | |
|------------------------------------|----|----|----|----|----|
| Length of tube (diameters)..... | 60 | 50 | 40 | 30 | 20 |
| Water per unit-weight of fuel..... | 12 | 11 | 10 | 9 | 8 |

When the ratio of heating to grate area was 25 to 1, Isherwood found the evaporation to vary thus:

| | | | | | | |
|---------------------|------|------|-----|-----|-----|-----|
| Fuel per hour... .. | 8 | 10 | 12 | 16 | 20 | 24 |
| Evaporation..... | 10.5 | 10.1 | 9.5 | 8.2 | 7.3 | 6.8 |

which series is represented by

$$W = \frac{21}{\sqrt[3]{F}}, \text{ nearly.}$$

Clark obtained with locomotives an equal evaporation with

| | | | | | | | | |
|-----------------------------|----|----|----|----|----|----|-----|-----|
| Fuel (coke)..... | 15 | 25 | 38 | 56 | 76 | 98 | 125 | 153 |
| Ratio of H. S. to G. S..... | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 |

the evaporation being constant at 9 of water to 1 of fuel, which may be expressed by

$$S = 8 \sqrt{F}, \text{ nearly,}$$

S being the ratio of the two areas and F the weight of coke burned on the unit of area of grate.

In estimating area of heating-surfaces the whole surface exposed to the hot-furnace gusts is reckoned. The formula for efficiency already given illustrates the progressive variation of the evaporative power with change of proportions of boiler.

The relation of size of boiler to quantity of steam demanded is one that occasionally becomes worthy of consideration. Where the steam is required for driving steam-engines it is very important that it should be thoroughly dry, and it is an advantage to moderately superheat it. Maximum economy cannot be attained where wet steam is used. A boiler attached to a steam-engine, and especially where fuel is costly and efficiency important, should have ample heating-surface, some superheating-surface if practicable, ample extent of water-surface area to permit free separation of steam and water, and large steam-space.

Steam employed for heating purposes is not necessarily dry; it may carry a large amount of water with it into the system of heating-coils or radiators, and yet give good results, if the latter are of large section. Where the pipes are of restricted area of section, however, wet steam flowing less freely than when dry or superheated, there may result such a retardation of flow and of circulation as may cause considerable increase of cost. This has been found sufficiently great, in some cases, to justify drying, and perhaps superheating, the exhaust-steam from engines where used for heating purposes. As a general rule, the boiler must be made a trifle larger to supply perfectly dry steam and do good work.

In the use of steam for heating purposes, one square foot of boiler-surface will supply from 7 to 10 square feet of radiating surface. Small boilers should be larger proportionately than large boilers. Each horse-power of boiler will supply from 250

to 350 feet of 1-in. steam-pipe, or 80 to 120 square feet of radiating surface.

Under ordinary conditions one horse-power will heat about—

| | |
|-----------------------------------------------|---------------------------|
| Brick dwellings, in blocks, as in cities..... | 15,000 to 20,000 cub. ft. |
| “ stores “ “ “ | 10,000 “ 15,000 “ “ |
| “ dwellings exposed all around..... | 10,000 “ 15,000 “ “ |
| “ mills, shops, factories, etc..... | 7,000 “ 10,000 “ “ |
| Wooden dwellings, exposed..... | 7,000 “ 10,000 “ “ |
| Foundries and wooden shops..... | 6,000 “ 10,000 “ “ |
| Exhibition buildings, largely glass, etc..... | 4,000 “ 15,000 “ “ |

The system of heating mills and manufactories by means of pipes placed overhead is recommended.

The air required for ventilation is usually warmed by the “indirect” system of radiation, the current passing through boxes or chambers in which a sufficient amount of pipe is coiled to heat it well. From 5 to 15 cubic feet per individual per minute are allowed, the former in crowded halls, the latter in dwellings, and about one-tenth as much for each gas-burner or lamp.

Small engines, according to Buel, demand steam, ordinarily, as below:

| Pressure of Steam in Boiler, by Gauge. | Pounds of Water per effective Horse- power per Hour. | Pressure of Steam in Boiler by Gauge. | Pounds of Water per effective Horse- power per Hour. |
|-------------------------------------------|------------------------------------------------------------|------------------------------------------|------------------------------------------------------------|
| 10 | 118 | 60 | 75 |
| 15 | 111 | 70 | 71 |
| 20 | 105 | 80 | 68 |
| 25 | 100 | 90 | 65 |
| 30 | 93 | 100 | 63 |
| 40 | 84 | 120 | 61 |
| 50 | 79 | 150 | 58 |

Pressures lower than 60 pounds are not usually adopted for small engines. Good examples of such engines have been found by the Author to demand from 25 to 33 per cent less steam, or feed-water, than is above given.

The table on the next page gives what are considered by the Author as fair estimates of water and steam consumption for the best classes of engines in common use, when of moderate size and in good order.

It is considered usually advisable to assume a set of practically attainable conditions in average good practice, and to take

NON-CONDENSING ENGINES.

| STEAM PRESSURE. | | POUNDS PER H. P. PER HOUR,—RATIO OF EXPANSION. | | | | | |
|-----------------|------------------|------------------------------------------------|----|----|----|----|----|
| Atmospheres. | Lbs. per sq. in. | 2 | 3 | 4 | 5 | 7 | 10 |
| 3 | 45 | 40 | 39 | 40 | 49 | 42 | 45 |
| 4 | 60 | 35 | 34 | 36 | 36 | 38 | 40 |
| 5 | 75 | 30 | 28 | 27 | 26 | 30 | 32 |
| 6 | 90 | 28 | 27 | 26 | 25 | 27 | 29 |
| 7 | 105 | 26 | 25 | 24 | 23 | 25 | 27 |
| 8 | 120 | 25 | 24 | 23 | 22 | 22 | 21 |
| 10 | 150 | 24 | 23 | 22 | 21 | 20 | 20 |

CONDENSING ENGINES.

| | | | | | | | |
|----|-----|----|----|----|----|----|----|
| 4 | 30 | 30 | 28 | 28 | 30 | 35 | 40 |
| 5 | 45 | 28 | 27 | 27 | 26 | 28 | 32 |
| 4 | 60 | 27 | 26 | 25 | 24 | 25 | 27 |
| 5 | 75 | 26 | 25 | 25 | 23 | 22 | 24 |
| 6 | 90 | 26 | 24 | 24 | 22 | 21 | 20 |
| 8 | 120 | 25 | 23 | 23 | 22 | 21 | 20 |
| 10 | 150 | 25 | 23 | 22 | 21 | 20 | 19 |

the power so obtainable as the measure of the power of the boiler in commercial and engineering transactions. The unit generally assumed has been usually the weight of steam demanded per horse-power per hour by a fairly good steam-engine. This magnitude has been gradually decreasing from the earliest period of the history of the steam-engine. In the time of Watt, one cubic foot of water per hour was thought fair; at the middle of the present century, ten pounds of coal was a usual figure, and five pounds, commonly equivalent to about forty pounds of feed-water evaporated, was allowed the best engines. After the introduction of the modern forms of engine this last figure was reduced twenty-five per cent, and the most recent improvements have still further lessened the consumption of fuel and of steam. By general consent, the unit has now become thirty pounds of dry steam per horse-power per hour, which represents the performance of good non-condensing mill engines.

16. General Schemes of Trial or Tests of Engines are adopted by engineers which, while varying in detail, all closely

resemble each other in their main purposes, and are somewhat similar in methods. They commonly include boiler-trials as the only practicable and satisfactory means of ascertaining the quantity and quality of steam supplied, and the cost of power in steam, fuel, and money. They invariably involve the application of the indicator or the dynamometer, and, if complete, of both, for power measurements. When the question to be solved is simply the efficiency of engine, or of engine considered dynamically, of the engine as a machine, a comparison of the indicated with the dynamometric power gives the solution; but when the thermal efficiency and the efficiency in transformation of energy is to be measured, the measurement of the quantity of energy supplied in the form of heat, and hence a boiler-trial, must necessarily form a part of the operation. All general systems may therefore be said to involve the whole series of determinations of quantity already indicated; but the details have not yet been authoritatively prescribed in such manner as to fix a standard system or standard methods. The experience of the most experienced and distinguished practitioners is, however, gradually producing a tolerably well settled custom in the more important of the several operations involved. Some such methods and some general plans of test-trials will be later described.

17. Steam-Boiler Trials, apart from the engine-test, and made for the purpose of ascertaining the quantity and quality of steam made, its cost in fuel and in combustible matter contained therein, and the efficiency of the boiler and of its heating-surface, are now very generally made by a fairly well recognized system. In the United States and in Germany, particularly, such methods are now made to follow very generally the prescribed order of procedure devised and published by engineers of recognized standing. Such a standard system is that proposed by the committee of the American Society of Mechanical Engineers, and this standard will be that accepted in this work.*

* Transactions American Society of Mechanical Engineers, vol. vii., 1884. A Manual of Steam-Boilers, by R. H. Thurston (N. Y. : J. Wiley & Sons, 1888), chap. xiv. pp. 484-537.

In the operation of conducting any trial, we have, usually, a single, well-understood object to attain, and the engineer should accustom himself to carefully define that object in his own mind, and to as carefully describe that object in his instructions and regulations for the proposed trial. The whole operation can then be carried on with that point distinctly in view, and the proposed end can then be accomplished with maximum economy of time and labor, as well as with greatest exactness. The observations must be made by the engineer conducting the trial, or by his assistants, with this object clearly in mind, and each should have a well-defined part of the work assigned him, and should assume responsibility for that part, having a distinct understanding in regard to the extent of his responsibility, and a good idea of the extent and nature of the work done by his colleagues, and the relations of each part to his own. No observations should be permitted to be made by unauthorized persons for entrance upon the log; and no duties should be permitted to be delegated by one assistant to another, without consultation and distinct understanding with the engineer in charge. The aim of the observers is, in boiler trials, for example, to obtain an exact determination of the weight of fuel used, its proportion of combustible matter effective in developing heat, the exact weight of water evaporated under the known conditions of the trial, into steam, the determination of the character of that steam, and often the nature of the combustion and the composition of the furnace-gases. Each of these distinct objects requires the determination of certain well-defined quantities, and the observer to whom each set of observations is intrusted should, whenever possible, be made sufficiently well acquainted with the object to be attained, and the method to be pursued in reaching it, to be able to make his own readings with accuracy, and to work up the results correctly. It is only after he has acquired this knowledge that he can be expected to do his work without direct supervision, and with satisfactory precision. The trial should, wherever possible, be so conducted that any error that may occur in the record may be detected, checked, or, if advisable, removed, by some process of mutual verification of

related observations. It is in this direction that the use of graphical methods of record and automatic instruments has greatest value. We should lose no opportunity to introduce both.

18. Engine Trials may or may not include determination of boiler performance and efficiency; but if they are to be satisfactorily complete, measurements of the quantity and quality of the steam supplied are as essential as any other determinations of quantity. In some cases, only a comparison of the work done with its cost in fuel is called for; but in this case the total efficiency so obtained cannot be analyzed into the two factors, engine efficiency and boiler efficiency, and it is impossible to say to what extent engine or boiler is responsible for the final results obtained. The complete investigation of the action and performance of the engine, as a heat-engine and prime motor, must always include some method of obtaining a measure of the amount of heat-energy supplied to the machine; the proportion of that energy which reaches the engine in available form; the distribution and disposition of the available part; the extent to which it is converted into useful work and into wasted power; the amount in detail of the various wastes; the method as well as extent of wastes; and the ameliorating or exaggerating effect of any observable accidental or purposely produced variations of condition and of operation upon the wastes, the economies, and the several efficiencies of the engine.

It is thus important that ways should be found and methods practised, that will determine the quantity and quality—whether wet or dry—of the steam supplied; the pressures and volumes of every stage of transfer and of transformation, and the quantities of heat stored, conveyed, and utilized or wasted.

19. Engine and Boiler Tests are thus necessarily commonly conducted simultaneously in the settlement of important contracts, and essential data can only be thus secured. Where the quantities to be ascertained and measured are not likely to vary greatly with period of operation, a trial of a few hours' duration will answer all purposes. Gas-engines are often tested a single hour, and five hours is quite as long as is often de-

sirable. Steam-engine and boiler trials are seldom less than ten hours in length, often occupy a full day of twenty-four hours, sometimes last a week, unintermittedly, night and day, and it is even sometimes prescribed that the more important data shall be recorded for several months or for a year at a time. Ordinarily, a ten-hour trial is quite sufficient, if properly conducted.

20. The Apparatus of Steam-boiler Trials consists of tanks to receive and in which to weigh the feed-water; scales with which to effect these measurements and to weigh fuel and ashes; thermometers with which to determine the temperature of the water and steam, and pyrometers for the furnace-flue and chimney temperatures; and it is now usual to employ a calorimeter with which to determine the condition of the steam, and to measure the proportion of entrained water. Before the systems of boiler trial usually adopted are employed, it will be necessary to understand the methods of use and of calibration and of standardizing these various kinds of apparatus, the sources from which they may be obtained, and the best methods of their application to the securing of the needed data. It is usually thought best to *weigh* all water, rather than measure its volume. If measured, it should be carefully noted that its variation of density with temperature is considerable, and sufficient to introduce observable errors if a constant density is assumed.

The Apparatus of Engine-testing consists of steam-engine indicators, dynamometers, counters and gauges, and good timing instruments. The use of these instruments and the methods of test and correction are simple; and, with the exception of the indicator, none demands very extended notice. The indicator, however, is an instrument which must be made with the utmost possible care and skill, and the study and interpretation of its diagrams is a matter demanding some skill, knowledge, and experience. Considerable space will therefore be devoted to the description of this interesting and indispensable instrument, its uses and applications, to the study of the methods of interpretation of its record, and necessary measurement and computations.

CHAPTER II.

STEAM-BOILER TRIALS.*

21. The Object of a Trial of a Steam-boiler is to determine what is the quantity of steam that a boiler can supply under definitely prescribed conditions; what is the quality, as to moisture or dryness, of that steam; what is the amount of fuel demanded to produce that steam; what the character of the combustion, and the actual conditions of operation of the boiler when at work. The conditions prescribed for one trial may differ greatly from those of another trial, and such differences are often the essential matters to be studied. In any case it is assumed that the conditions under which the boiler is to be worked are to be definitely stated, and the engineer conducting the experiments is expected to ascertain all the facts which go to determine the performance of the boiler, and to state them with accuracy, conciseness, and completeness.

In the attempt to ascertain those facts the engineer meets with some difficulties, and finds it necessary to exercise the utmost care and skill. In conducting a steam-boiler trial the weight of the water supplied to the boiler must be determined; the weight of the fuel consumed must be obtained; the state of the steam made must be determined; and these quantities must all be noted at frequent intervals. It is also necessary to know whether the combustion is perfect or imperfect, and to what extent the conditions and facts noted are due to the boiler, and what to external conditions.

It has now come to be considered that the determination of power and economy of a steam-boiler demands all the care, skill, and perfection of method and of apparatus of any purely scientific investigation. It is desirable that all work of this kind shall be done in substantially the same way, in order that comparisons may be made.

* Mainly from the Author's "Manual of Steam Boilers," New York; published by John Wiley & Sons.

22. Tests of Value of Fuel are sometimes the sole object of a trial of a steam-boiler, the intent being to ascertain by actual experiment what quantity of water a fuel of unknown quality can evaporate in a boiler of which the general efficiency is fairly well established. In such cases the fuel is employed in the usual manner and the results compared with those obtained with fuels of known excellence. Thus, in a good type of boiler, having a good proportion of area of heating-surface to weight of fuel burned per hour, it may be found that a fuel of established reputation for uniform excellence will evaporate ten times its own weight of water "from and at" the boiling-point. The trial of a fuel of unknown quality may prove that this boiler will, under precisely similar conditions, evaporate an equal amount of water into steam, and yet the market price of the fuel may be considerably less than that of the other. The immediate result would be the substitution of the second for the first, should no counterbalancing disadvantages exist. In such cases the method of conducting the experiment is precisely the same as where the efficiency of the boiler is determined; but the object sought is quite a different one. This also commonly compels at least two trials, the one of the old and standard, the other of the new and uncertain fuel, and a comparison of boiler-efficiency as found in the two trials.

23. The Determination of the Value of a steam-boiler involves the measurement of its efficiency, independently of the nature of the fuel, and it is thus important that a standard system of measuring the effectiveness of the fuel should be settled upon, or that all variations of such effectiveness should be eliminated. The latter is commonly the course taken; and the determination of the efficiency of the boiler is based upon the measurement of the evaporation of water, under stated standard conditions, per unit weight of the combustible and burned portion of the fuel supplied during the trial.

But the power of the boiler is as important an element of its value as its efficiency, and a complete trial includes, usually, measurements of efficiency at both the rated and the maximum working power of the boiler as operated for its special purpose.

24. The Evaporative Power of Fuels depends upon

not only their chemical composition as fuels, but also to an important extent upon their structure and their physical condition in every aspect; on their greater or less purity, and the admixture of earths, moisture, or other foreign matters; the fitness of the furnace for their utilization; the air-supply; its quantity, temperature, and humidity; the proximity of chilling surfaces; the extent of the combustion-chamber in which the gases rising from the bed of coal or other combustible may be more or less completely consumed; and many other minor conditions, all of which tell, in a more or less important degree, upon their value and the efficiency of the system of heat-generation.

25. Analyses of Fuels are sometimes made, either as a check upon the results of the trial or in substitution for it. Should analysis show that a given fuel is rich in heat-producing elements, while trial fails to give the results that should have been obtained, and such as the use of other fuels in the same boilers indicates to be possible, it will at once appear that the fuel demands peculiar treatment, or some other arrangement of furnace. Should doubt exist which of a number of fuels of the same class is best, chemical analysis may give a quicker and cheaper answer to the question than a formal trial. It rarely happens, however, that any system is as satisfactory, in the end, as actual trial extending over so long a period as to eliminate uncertainties.

Methods of analysis differ somewhat. The following is a standard method of general treatment as prescribed by the Union of Engineers of Germany:*

In order to take a sample of the fuel, a shovelful from each barrow or wagon will be thrown into a box with a cover. The coal will be mixed up and spread in the form of a square upon a level floor, and then divided by two diagonals into four parts. Of these, two opposite parts will be taken away, the other two will be broken up small and mixed together. Another shovelful will then be thrown in, and the method continued until about 10 kilogrammes are in the box. This will then be

* *American Engineer*, August, 1883.

closed and reserved for chemical analysis. For accurate experiments the halves which have been taken away should also be analyzed.

To determine the moisture in the coal, about 10 grammes from the above-named sample is to be heated for two hours to 105° or 110° C. The loss in weight shows the moisture in the coal. Coal which happens to have been wetted by rain or otherwise should not be used. The test should be applied to coal in the average state of moisture at which it is delivered from the pit mouth, and this state should, if necessary, be determined beforehand. The remainder of the sample, powdered and mixed thoroughly, serves to determine the ash, the carbon, the hydrogen, the nitrogen, and the sulphur. The heating-value of the coal is determined as follows: Suppose that it is found to contain c per cent of carbon, h per cent of hydrogen, s per cent of sulphur, o per cent of oxygen, and w per cent of water, then the theoretical heating-value is given by the formula of Dulong as follows:

(a). *Referred to Water at 0° Cent.*

$$8100c + 34320 \left(h - \frac{o}{8} \right) + 2500s.$$

(b). *Referred to Water at 100° Cent.*

$$8100c + 34200 \left(h - \frac{o}{8} \right) + 2500s - 636.5 (9h + w.)$$

To determine the quantity of air required for burning coal we have the following: One kilogramme of coal requires to burn it,

$$\frac{2.667c + 8h + s - o}{100 \times 1.43} \text{ cu. metres of oxygen ; or,}$$

$$\frac{2.667c + 8h + s - o}{21 \times 1.43} \text{ cu. metres of air containing 21 per ct. of oxygen.}$$

The analyses should be made with care, by a skilled and experienced chemist, if any important question is to be settled.

26. Economy of Fuel is nearly synonymous with efficiency of boiler, as a matter of engineering simply; but when the finance of the case is studied, it is often found, from that

point of view, a very different matter. It is perfectly possible to adopt so great a proportion of heating-surface, so large a boiler, that the gain in fuel saved, as compared with boilers of similar type and usual proportions, may be more than offset by the increased charges on account of enlargement of boiler. The efficiency of boiler, in the ordinary sense in which that term is used, is, however, a measure of economy. The variation of efficiency and of economy in fuel consumption is a function of the proportion of area and of heating-surface to fuel burned, and the object of a boiler-trial is to ascertain these relations with precision. An understanding should be had before the trial in regard to the kind of fuel to be used; where no reason of controlling importance exists to the contrary, the best obtainable coal should be selected, for the reason that a boiler can be better judged, and the results of its trial may be more satisfactorily compared with similar trials of other boilers, when the very best work of which it is capable is done by it. The differences between separate lots of the best coals are less than the differences between separate lots of inferior fuels, and the comparison is thus less difficult where the former are used.

27. The Relative Values of Boilers depend not only on their efficiencies, but also on their capacities for furnishing steam, and on various other qualities and attributes: as their greater or less complication in structure; their safety and durability; their volume, weight, and cost. The boiler-trial only settles questions relating to their efficiency and capacity, and their real relations of value, only just so far as those elements enter the problem. These are usually, however, the main factors, and their measurement by a test-trial gives the means of deciding, in nearly all cases, every question likely to present itself in the use of the apparatus.

28. Variations of Efficiency occur with variations in grate-area, in rate of combustion and in kind of fuel. In any given boiler, within a wide range of which the limits are usually far outside of practical conditions, the greater the quantity of fuel burned the less the amount of steam made per unit weight of that fuel; the smaller the quantity of fuel,

burned under proper conditions, in the boiler, the higher the efficiency; and it has been seen in an earlier chapter, that the gain in efficiency, with increasing proportion of heating to grate surface or to fuel burned, is less and less as this increase goes on. By enlarging or reducing the grate, or by increasing or diminishing the draught and air-supply, and during a succession of trials, noting the method of variation of efficiency and of capacity for making steam, the law of such variations may be established, and the best arrangement, all things considered, may be determined.

29. Variations of Proportions in different boilers, otherwise similar, have been seen to be capable of expression by a very simple algebraic expression on which all theories of efficiency are based. But in some cases this law is not found to be precisely applicable, and only test-trials of boilers so differing can be relied upon to give correct relations. The general relations already stated invariably hold; but it often happens that a steam-boiler exhibits peculiarities which make that exact statement inapplicable. It is not uncommon not only to compare actual performance, as shown by trial, with the results indicated by the theory, but also to alter the ratio of heating to grate surface by bricking over more or less of the grate, and by this or other expedients so varying that ratio in successive trials as to obtain an empirical and approximately exact expression for the law of variation of efficiency for the particular case in hand.

30. Combined Power and Efficiency distinguish the best types of boiler. That which, at a given cost, exhibits highest steam-producing power combined with greatest efficiency, is the best boiler. These qualities; however, are not usually compatible, and increased steam-production from any boiler is commonly attended with a decrease in efficiency; and as the one or the other of these qualities is the more important, the combination which will give best total result will vary. In no two cases will the same combination be equally desirable. Every boiler must be tested for both before it can be said whether it is satisfactorily adapted to its place and work.

31. The Apparatus and Methods of test-trials should be

prescribed in the preliminary arrangements for every trial, and if possible should be in exact accordance with some accepted standard rules. The apparatus consists of scales and tanks for measurement of weights of coal and of water; gauges to give the pressure of steam; thermometers of great accuracy to determine the temperatures of water, steam, and flue-gases; and calorimeters to determine the quality of the steam and the extent of superheating, or the percentage of moisture entrained by it.

The establishment of the correctness of this apparatus is the first of the preliminaries to their use. The standardization of the instruments is a matter of supreme importance, since upon their accuracy the whole work of the engineer is dependent. It is also a work demanding, in most cases, unusual skill and care, and, to be satisfactory, must generally be performed either at the manufacturer's, or at the office of the engineer conducting the trial. The scales can usually be standardized by the official sealer of weights and measures, and sealed by him; the water-meters, if used, can be readily tested by the use of the scales so sealed; the thermometers are, as a rule, best tested by their makers, and should be sent to the maker for test immediately before and directly after the test. The engineer often has a carefully preserved standard with which they may be compared in his own office. The same remarks apply to the examination of the gauges used, which should be standardized both before and after their use. The apparatus used in connection with the calorimeter, in the determination of the quality of the steam made, demand exceptional care in this process. Where it is unavoidable, the use of coarsely graduated thermometers and roughly constructed scales may be permitted, but only then when a very large number of observations are taken, and an average thus obtained which may be fairly expected to fall within reasonable limits of error.

The method of starting and of stopping the trial is a very important matter, and one upon which engineers of experience and acknowledged authority are not in complete accord. The principles to be adhered to in this matter, as in every other detail of the operation of testing a boiler, are easily specified,

but they are not always as easy of practice. All conditions should be as exactly the same at the beginning and at the end of the test as they can possibly be made. The period of the trial and the times of stopping and of starting should be capable of being exactly fixed, and the method of test should be such as should permit of the commencement and the end occurring at these exactly defined times, or, as an alternative, they should be such that the work done by the boiler during the less precisely determinable time of beginning and ending of the trial should be as nearly as possible *nil*, so that a slight error as to time may not appreciably affect the results.

During the trial, provision should be made for the preservation of the utmost possible uniformity of working conditions throughout the whole period of the trial. Every irregularity gives rise to more or less loss of efficiency, and to uncertainty in regard to the correctness of the reported figures. The nearer the working of the boiler is kept to the final average for the trial, the better.

Uniformity of operation and maximum efficiency are best attainable during a trial when a system of record is adopted which allows of that regularity being shown at all times; and records in proper form are the best possible security against error of observation. Graphical methods should be adopted wherever practicable. Such methods of record exhibit most satisfactorily the accordance with or the deviation from the uniformity of operation considered so desirable on the score of efficiency and accuracy.

32. Standard Test-trials are made under established systems, and in accordance with codes of regulations which are accepted as representing a satisfactory system of procedure. In such cases the first step is to settle upon a standard of measurement and comparison that may be accepted by all who may be interested in the result. The standard nominal horsepower has already been described as now accepted by the best authorities.

The Committee of Judges of the Centennial Exhibition, to whom the trials of competing boilers at that exhibition were intrusted, adopted the unit, *30 pounds of water evaporated into*

dry steam per hour from feed-water at 100° Fahrenheit, and under a pressure of seventy pounds per square inch above the atmosphere, these conditions being considered to represent fairly average practice The quantity of heat demanded to evaporate a pound of water under these conditions is 1110.2 British thermal units, or 1.1496 "units of evaporation." The unit of power proposed is thus equivalent to the development of 33,305 heat-units per hour, or 34.488 units of evaporation. The "unit of evaporation" is taken as a certain weight—preferably unity of water, evaporated "from and at" the boiling-point under atmospheric pressure. The now-accepted unit of boiler-power, in the code constructed for the American Society of Mechanical Engineers,* is the equivalent of the Centennial Standard, and in all standard trials the commercial horse-power is taken as *an evaporation of 30 pounds of water per hour from a feed-water temperature of 100° Fahr. into steam at 70 pounds gauge-pressure*, which is equal to $34\frac{1}{2}$ units of evaporation, that is, to $34\frac{1}{2}$ pounds of water evaporated from a feed-water temperature of 212° Fahr. into steam at the same temperature. This standard is taken as the equivalent of 33,305 thermal units per hour.†

A boiler rated at any stated horse-power should be capable of developing that power with easy firing, moderate draught and ordinary fuel, while exhibiting good economy; and the boiler should be capable of developing one half or one third more than its rated power to meet emergencies at times when maximum economy is not the most important object to be attained.

33. Instructions and Rules governing the standard system of boiler-trial, prepared by a committee of the American Society of Mechanical Engineers, may be taken as a good illustration of such regulations as, in one form or another, have

* Transactions, vol. vi., 1884.

† An evaporation of 30 pounds of water from 100° F. into steam at 70 pounds pressure is equal to an evaporation of 34.488 pounds from and at 212°; and an evaporation of $34\frac{1}{2}$ pounds from and at 212° F. is equal to 30.010 pounds from 100° F., into steam at 70 pounds pressure.

The "unit of evaporation" being taken equal to 965.7 thermal units, the commercial horse-power is $34.488 \times 965.7 = 33.305$ thermal units.

been customarily agreed upon by engineers conducting such work. They are as follows:

PRELIMINARIES TO A TEST.

I. *In preparing for* and conducting trials of steam-boilers, the specific object of the proposed trial should be clearly defined and steadily kept in view.

II. *Measure and record the dimensions*, position, etc., of grate and heating surfaces, flues and chimneys, proportion of air-space in the grate-surface, kind of draught, natural or forced.

III. *Put the Boiler in good condition*.—Have heating-surface clean inside and out, grate-bars and sides of furnace free from clinkers, dust and ashes removed from back connections, leaks in masonry stopped, and all obstructions to draught removed. See that the damper will open to full extent, and that it may be closed when desired. Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will then escape through the leaks.

IV. *Have an understanding with the parties* in whose interest the test is to be made as to the character of the coal to be used. The coal must be dry, or, if wet, a sample must be dried carefully and a determination of the amount of moisture in the coal made, and the calculation of the results of the test corrected accordingly.

Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Alleghany Mountains good anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Alleghany Mountains and east of the Missouri River, Pittsburg lump coal may be used.*

V. *In all important tests* a sample of coal should be selected for chemical analysis.

* These coals are selected because they are almost the only coals which contain the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

VI. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are :

1. Scales for weighing coal, ashes, and water.
2. Tanks, or water-meters for measuring water. Water-meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank.
3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc.
4. Pressure-gauges, draught-gauges, etc.

VII. *Before beginning a test*, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new, it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

VIII. Before beginning a test, the boiler and connections should be free from leaks, and all water-connections, including blow and extra-feed pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which water is to be fed to the boiler during the trial. In locations where the reliability of the power is so important that an extra feed-pipe must be kept in position, and in general when for any other reason water-pipes other than the feed-pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks, or accidental or unauthorized opening of valves. During the test the blow-off pipe should remain exposed.

If an injector is used, it must receive steam directly from the boiler being tested, and not from a steam-pipe, or from any other boiler.

See that the steam-pipe is so arranged that water of condensation cannot run back into the boiler. If the steam-pipe has such an inclination that the water of condensation from any portion of the steam-pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

STARTING AND STOPPING A TEST.

A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam-pressure should be the same, the water-level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact uniformity as possible in conditions of the fire and in temperatures of the walls and flues, the following method of starting and stopping a test should be adopted:

X. *Standard Method*.—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water-level while the water is in a quiescent state, just before lighting the fire.

At the end of the test, remove the whole fire, clean the grates and ash-pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water-level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating pump after test is completed. It will generally be necessary to regulate the discharge of steam from the boiler tested by means of the stop-valve for a time while fires are being hauled at the beginning and at the end of the test, in order to keep the steam-pressure in the boiler at those times up to the average during the test.

XI. *Alternate Method*.—Instead of the Standard Method above described, the following may be employed where local conditions render it necessary:

At the regular time for slicing and cleaning fires have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of

steam and the height of the water-level—which should be at the medium height to be carried throughout the test—at the same time ; and note this time as the time of starting the test. Fresh coal, which has been weighed, should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as at the start. The water-level and steam-pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

DURING THE TEST.

XII. *Keep the Conditions Uniform.*—The boiler should be run continuously, without stopping for meal-times or for rise or fall of pressure of steam due to change of demand for steam. The draught being adjusted to the rate of evaporation or combustion desired before the test is begun, it should be retained constant during the test by means of the damper.

If the boiler is not connected to the same steam-pipe with other boilers, an extra outlet for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety-valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to a main steam-pipe with other boilers, the safety-valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as force of draught, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion, and the kind of grates. When very good coal is used, and the combustion not too rapid, a ten-hour test may be run without any cleaning

of the grates, other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

XIII. *Keeping the Records.*—The coal should be weighed and delivered to the firemen in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

XIV. *Priming Tests.*—In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent, and the final records of the boiler test corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate to within a tenth of a degree, and the scales on which the water is weighed to within one hundredth of a pound.

ANALYSES OF GASES.—MEASUREMENT OF AIR-SUPPLY, ETC.

XV. In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general not necessary in tests for commercial purposes. These are the measurement of the air-supply, the determination of its con-

REPORTING THE TRIAL.

XVII. The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adapted for the specific object for which the trial is made. The items marked with a * may be omitted for ordinary trials, but are desirable for comparison with similar data from other sources.

Results of the trials of a.....
 Boiler at.....
 To determine.....

| | | | |
|--------------------------------------------------------------------------------|-----------|--|--|
| | | | |
| 1. Date of trial..... | hours. | | |
| 2. Duration of trial..... | | | |
| DIMENSIONS AND PROPORTIONS. | | | |
| Leave space for complete description. See Appendix XXIII. | | | |
| 3. Grate surface....wide....long....Area.... | sq. ft. | | |
| 4. Water-heating surface..... | sq. ft. | | |
| 5. Superheating-surface..... | sq. ft. | | |
| 6. Ratio of water-heating surface to grate-surface..... | | | |
| AVERAGE PRESSURES. | | | |
| 7. Steam-pressure in boiler, by gauge..... | lbs. | | |
| *8. Absolute steam-pressure | lbs. | | |
| *9. Atmospheric pressure, per barometer..... | in. | | |
| 10. Force of draught in inches of water..... | in. | | |
| AVERAGE TEMPERATURES. | | | |
| *11. Of external air..... | deg. | | |
| *12. Of fire-room..... | deg. | | |
| *13. Of steam..... | deg. | | |
| 14. Of escaping gases..... | deg. | | |
| 15. Of feed-water..... | deg. | | |
| FUEL. | | | |
| 16. Total amount of coal consumed †..... | lbs. | | |
| 17. Moisture in coal..... | per cent. | | |
| 18. Dry coal consumed..... | lbs. | | |
| 19. Total refuse, dry..... pounds =..... | per cent. | | |
| 20. Total combustible (dry weight of coal, Item 18, less refuse, Item 19)..... | lbs. | | |
| *21. Dry coal consumed per hour..... | lbs. | | |
| *22. Combustible consumed per hour..... | lbs. | | |

* See reference in paragraph preceding table.

† Including equivalent of wood used in lighting fire. 1 pound of wood equals 0.4 pound coal. Not including unburnt coal withdrawn from fire at end of test.

| RESULTS OF CALORIMETRIC TESTS. | | | |
|--------------------------------------------------------------------------------------------------|-----------|--|--|
| 23. Quality of steam, dry steam being taken as unity..... | | | |
| 24. Percentage of moisture in steam..... | per cent. | | |
| 25. Number of degrees superheated..... | deg. | | |
| WATER. | | | |
| 26. Total weight of water pumped into boiler and apparently evaporated*..... | lbs. | | |
| 27. Water actually evaporated, corrected for quality of steam †..... | lbs. | | |
| 28. Equivalent water evaporated into dry steam from and at 212° F. †..... | lbs. | | |
| *29. Equivalent total heat derived from fuel in British thermal units †..... | B. T. U. | | |
| 30. Equivalent water evaporated into dry steam from and at 212° F. per hour..... | lbs. | | |
| ECONOMIC EVAPORATION. | | | |
| 31. Water actually evaporated per pound of dry coal, from actual pressure and temperature †..... | lbs. | | |
| 32. Equivalent water evaporated per pound of dry coal from and at 212° F. †..... | lbs. | | |
| 33. Equivalent water evaporated per pound of combustible from and at 212° F. †..... | lbs. | | |

* Corrected for inequality of water-level and of steam-pressure at beginning and end of test.

† The following shows how some of the items in the above table are derived from others:

Item 27 = Item 26 × Item 23.

Item 28 = Item 27 × Factor of evaporation.

Factor of evaporation = $\frac{H - h}{965.7}$, H and h being respectively the total heat-units in steam of the average observed pressure and in water of the average observed temperature of feed, as obtained from tables of the properties of steam and water.

Item 29 = Item 27 × ($H - h$).

Item 31 = Item 27 ÷ Item 18.

Item 32 = Item 28 ÷ Item 18 or = Item 31 × Factor of evaporation.

Item 33 = Item 28 ÷ Item 20 or = Item 32 ÷ (per cent 100 - Item 19).

Items 36 to 38. First term = Item 20 × $\frac{6}{5}$

Items 40 to 42. First term = Item 39 × 0.8698.

Item 43 = Item 29 × 0.00003 or = $\frac{\text{Item } 30}{34\frac{1}{2}}$.

Item 45 = $\frac{\text{Difference of Items 43 and 44}}{\text{Item 44}}$.

| COMMERCIAL EVAPORATION. | | | | |
|-------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----------|--|--|
| 34. | Equivalent water evaporated per pound of dry coal with one sixth refuse, at 70 pounds gauge-pressure, from temperature of 100° F. = Item 33 multiplied by 0.7249. | lbs. | | |
| RATE OF COMBUSTION. | | | | |
| 35. | Dry coal actually burned per square foot of grate-surface per hour. | lbs. | | |
| *36. | Consumption of dry coal per hour. | lbs. | | |
| *37. | Coal assumed with one sixth refuse.† | lbs. | | |
| *38. | Per sq. ft. of grate-surface. | lbs. | | |
| | Per sq. ft. of water-heating surface.... | lbs. | | |
| | Per sq. ft. of least area for draught... | lbs. | | |
| RATE OF EVAPORATION. | | | | |
| 39. | Water evaporated from and at 212° F. per square foot of heating-surface per hour... | lbs. | | |
| *40. | Water evaporated per hour from temperature of 100° F. | lbs. | | |
| *41. | Per sq. ft. of water-heating surface.. | lbs. | | |
| *42. | into steam of 70 pounds gauge-pressure.† | lbs. | | |
| | Per sq. ft. of least area for draught. | lbs. | | |
| COMMERCIAL HORSE-POWER. | | | | |
| 43. | On basis of thirty pounds of water per hour evaporated from temperature of 100° F. into steam of 70 pounds gauge pressure, (= 34½ lbs. from and at 212°)†..... | H. P. | | |
| 44. | Horse-power, builders' rating, at....square feet per horse-power. | H. P. | | |
| 45. | Per cent developed above, or below, rating†..... | Per cent. | | |

34. **Precautions** are to be taken in every possible way to prevent and avoid irregularities in the conduct of the trial and errors of observation.*

In preparing for and conducting trials of steam-boilers the specific object of the proposed trial should be clearly defined and steadily kept in view, and as suggested by Mr. Hoadley—

(1) If it be to determine the efficiency of a given style of boiler or of boiler-setting under normal conditions, the boiler brickwork, grates, dampers, flues, pipes, in short, the whole apparatus, should be carefully examined and accurately described,

* The appendix to the report above quoted should be read in this connection.

and any variation from a normal condition should be remedied, if possible, and if irremediable, clearly described and pointed out.

(2) If it be to ascertain the condition of a given boiler or set of boilers with a view to the improvement of whatever may be faulty, the conditions actually existing should be accurately observed and clearly described.

(3) If the object be to determine the relative value of two or more kinds of coal, or the actual value of any kind, exact equality of conditions should be maintained if possible, or, where that is not practicable, all variations should be duly allowed for.

(4) Only one variable should be allowed to enter into the problem; or, since the entire exclusion of disturbing variations cannot usually be effected, they should be kept as closely as possible within narrow limits, and allowed for with all possible accuracy.

Blanks should be provided in advance, in which to enter all data observed during the test. The preceding instructions contain the form used in presenting the general results. Records should be, as far as possible, made in a standard form, in order that all may be comparable.

The observations must be made by the engineer conducting the trial, or by his assistants, with this object distinctly in mind; and each should have a well-defined part of the work assigned him, and should assume responsibility for that part, having a distinct understanding in regard to the extent of his responsibility, and a good idea of the extent and nature of the work done by his colleagues, and the relations of each part to his own. No observations should be permitted to be made by unauthorized persons for entrance upon the log; and no duties should be permitted to be delegated by one assistant to another, without consultation and distinct understanding with the engineer in charge. The trial should, wherever possible, be so conducted that any error that may occur in the record may be detected, checked, or, if advisable, removed, by some process of mutual verification of related observations. It is in this direction that the use of graphical methods of record and automatic instruments have greatest value.

Several methods of weighing fuel have been found very satisfactory, but it should be an essential feature that the weights shall be made by one observer and checked by another, at as distant a point as is convenient. The weighing of the fuel by one observer at the point of storage, and the record at that point of times of delivery, as well as of weights of each lot, and the tallying of the number and record of the time of receipt at the furnace-door, will be usually found a safe system. The failure to record any one weight leads to similar error, and can only be certainly prevented by an effective method of double observation and check.

The same remarks apply, to a considerable extent, to the weighing of the water fed to the boiler. A careful arrangement of weighing apparatus, a double set of observations, where possible, and thus safe checks on the figures obtained, are essential to certainty of results. With good observers at the tank, and with small demand for water, a single tank can be used; but two are preferable in all cases, and three should be used if the work demands very large amounts of feed-water, as at trials of very large boilers, or of "batteries." The more uniform the water-supply, as well as the more steady the firing, the less the liability to mistake in making the record.

The two blanks which follow were prepared by the Author for use in laboratory as well as professional work.

Such blanks are always desirable, and are sometimes even more elaborate than those here illustrated.* For ordinary purposes, a less complete tabulation may be sufficient; while, for special purposes and for scientific investigations and the work of research, considerable more elaboration may be found desirable, or even necessary.

Graphical methods of representation of the data and conditions and of the progress of the trial, as elsewhere illustrated, are also often found exceedingly useful and convenient.

* *Vide* "Stevens Indicator," Nov. '89.

Trial made at _____
 on _____
 Fuel _____
 Composition _____

| DATE OF TRIAL. | LENGTH OF TRIAL. | AREAS. | | | | RATIO OF GRATE TO HEATING-SURFACE. | AVERAGE TEMPERATURES. | | | | AVERAGE PRESSURES. | | | | CONSUMPTION OF FUEL. | | | | ASHES. | | REMARKS. |
|------------------|------------------|---------|------------------|------------------------|-------------------------------|------------------------------------|-----------------------|-----------|----------------------|-------------|--------------------|--------------|----------------|--------|------------------------------------|------------------------------------------|--------|---------------------------|--------------------|--|----------|
| | | Grate. | Heating-surface. | Super-heating-surface. | Least Cross-section of flues. | | Boiler-room. | External. | Entrance to Chimney. | Feed-water. | Barometer. | Steam-gauge. | Draught-gauge. | Total. | Per square foot of grate per hour. | Per sq. ft. of heating-surface per hour. | Total. | Proportion of Total Fuel. | Total lbs. | | |
| Number of Trial. | Hours. | sq. ft. | sq. ft. | sq. ft. | sq. ft. | Fahr. | Fahr. | Fahr. | Fahr. | ins. | lbs. | ins. | lbs. | lbs. | lbs. | lbs. | perct. | lbs. | Total COMBUSTIBLE. | | |

| APPARENT EVAPORATION. | | | | | | | | | | REAL EVAPORATION. | | | | | | | | | | | |
|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|--|-------------------------------------------------------------------------------|
| Per Pound of Fuel. | Per Pound of Combustible. | | | | Per Square Foot of Heating-surface, per Hour. | | | | Per Pound of Fuel. | | | | Per Pound of Combustible. | | | | Per Square Foot of Heating-surface, per Hour. | | | | Square feet of Heating-surface required to Evaporate one Cubic Foot of Water. |
| | lbs. | lbs. | lbs. | lbs. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | | |
| From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | | |

TABLE II.—Continued.

| TOTAL WATER FED TO BOILER. | | | | AVERAGE PRIMING | TOTAL WATER PRIMED. | WATER EVAPORATED INTO DRY STEAM. | | | | REMARKS. |
|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-----------------|---------------------|---------------------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|-------------------------------------------------------|----------|
| From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | | | From actual temperature of feed-water and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | Equivalent from 212° F. and at actual steam-pressure. | |
| lbs. | lbs. | lbs. | lbs. | per cent. | lbs. | lbs. | lbs. | lbs. | lbs. | |

| Average Amount of Superheating. | EVAPORATION FROM AND AT 212° F., EQUIVALENT TO TOTAL HEAT-UNITS DERIVED FROM FUEL. | | | | EFFICIENCY. | | VALUES OF A AND B IN $F = A \sqrt{H} + B$. | HORSE-POWER. | | REMARKS. |
|---------------------------------|------------------------------------------------------------------------------------|---------------------------|------------------------------------------|-----------|---------------|------------|---------------------------------------------|--------------|---------|----------|
| | Per Pound of Fuel. | Per Pound of Combustible. | Per sq. ft. of Heating-surface per hour. | Per cent. | Experimental. | Estimated. | | Rated. | Actual. | |
| Fahr. | lbs. | lbs. | lbs. | per cent. | per cent. | per cent. | | | | |

The following is a good form of simple, but usually sufficient, boiler-room log, for everyday purposes : *

|Mo.....188 .. | HOURS. | | | | | | | | | | | Total. | Average. |
|-------------------------|--------|---|---|---|----|----|----|---|---|---|---|--------|----------|
| | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 1 | 2 | 3 | 4 | 5 | |
| B. Press. by gauge..... | | | | | | | | | | | | | |
| Fuel Fired, lbs..... | | | | | | | | | | | | | |
| Ashes and uncons..... | | | | | | | | | | | | | |
| Combustible, lbs..... | | | | | | | | | | | | | |
| Uptake temp..... | | | | | | | | | | | | | |
| Feed temp..... | | | | | | | | | | | | | |
| | | | | | | | | | | | | | |
| Turns per min..... | | | | | | | | | | | | | |
| Throttle open..... | | | | | | | | | | | | | |
| Av. Cut-off..... | | | | | | | | | | | | | |
| Av. Vacuum, in..... | | | | | | | | | | | | | |
| Hot-well temp..... | | | | | | | | | | | | | |
| Injection temp..... | | | | | | | | | | | | | |
| " lbs..... | | | | | | | | | | | | | |
| Indicated H. P..... | | | | | | | | | | | | | |
| | | | | | | | | | | | | | |

Total hours run.....
 Kind of coal.....
 Cost coal per 2,240 lbs.....
 Oil, gals.....
 Kind of Oil.....
 Cost of oil per gal.....
 Waste, lbs.....

Steam, dry or wet.....
 Lbs. water per lb. coal.....
 Lbs. water per lb. combustible.....
 Lbs. " " coal from and at 212°..
 Lbs. " " comb. from and at 212°..
 Remarks :

35. The Heating Power of any Fuel is determined by calculating its *total heat of combustion*. This quantity is the sum of the amounts of heat generated by the combustion of the unoxidized carbon and hydrogen contained in the fuel, less the heat required in the evaporation and volatilization of constituents which become gaseous at the temperature resulting from the combustion of the first-named elements. It is measured in "thermal units."

A *thermal unit* is the quantity of heat necessary to raise a unit weight of water, at temperature of maximum density, one degree of temperature. The British thermal unit is the quantity of heat required to raise a pound of water from the temperature 39°.1 to 40°.1 Fahr. The metric unit or calorie is the

* See The Practical World, Dec. 1888, p. 1.

quantity of heat required to raise one kilogramme of water (2.2046215 pounds) from $3^{\circ}.94$ to $4^{\circ}.94$ Centigrade.

One metric or Centigrade unit is equal to 3.96832 British units, and a British unit is equal to 0.251996 metric unit.

An approximate estimate of the number of thermal units developed by the combustion of a pound or kilogramme of any dry fuel, of which the chemical composition is known, may be obtained by the use of the following formula :

$$\left. \begin{aligned} h &= 14,500C + 62,000\left(H - \frac{O}{8}\right), \\ h' &= 8,080C + 34,460\left(H - \frac{O}{8}\right), \end{aligned} \right\}$$

where h is the number of British thermal units representing the total heat of combustion of one pound of the fuel; h' is the number of metric units per kilogramme of fuel; C represents the percentage of carbon; H that of hydrogen; and O that of oxygen.

Thus an anthracite coal has been found to have the following composition :

COMPOSITION OF ANTHRACITE COAL.

| | Per cent. |
|--------------------------------------|-----------|
| Carbon..... | 81.34 |
| Hydrogen <i>uncombined</i> | 3.45 |
| Hydrogen <i>in combination</i> | 0.74 |
| Oxygen and Nitrogen..... | 5.89 |
| Sulphur..... | 0.64 |
| Water..... | 2.00 |
| Ash..... | 5.94 |
| Total..... | 100.00 |

One pound or kilogramme of coal, of which the above is an analysis, can evaporate theoretically 14.4 pounds or kilogrammes of water from and at 100° Centigrade, or 212° Fahr.

MM. Scheurer, Kestner, and Meunier have adopted the common formula as first proposed by Dulong, but would omit

all account of oxygen, thus reducing, as is claimed, the average error of the formula from about 12 per cent or more to 8 or 10. M. Cornut would separate the fixed from the volatile carbon, and would give the latter about one third more credit for heating power than the former ; closer approximations are thus made than by the other methods.

Mr. G. H. Babcock gives the following tables as representative of familiar practice :

| KIND OF COMBUSTIBLE. | AIR REQUIRED. | TEMPERATURE OF COMBUSTION. | | | | THEORETICAL VALUE. | | HIGHEST ATTAINABLE VALUE UNDER BOILER. | |
|----------------------------------|-----------------------------------|---------------------------------|----------------------------------------------------------|-------------------------------------------|-------------------------------------------------|----------------------------------------------------|-----------------------------------------------------------------------|----------------------------------------|---------------------------------------------------------|
| | In lbs. per pound of combustible. | With Theoretical Supply of Air. | With $1\frac{1}{2}$ times the theoretical supply of air. | With twice the theoretical supply of air. | With three times the theoretical supply of air. | In lbs. of water raised 1° per lb. of combustible. | In lbs. of water evaporated from and at 212°, with 1 lb. combustible. | With chimney draught. | With blast, theoretical supply of air at 60°, gas 320°. |
| Hydrogen..... | 36.00 | 5,730 | 3,860 | 2,860 | 1,940 | 62,032 | 64.20 | | |
| Petroleum..... | 15.43 | 5,050 | 3,515 | 2,710 | 1,850 | 21,000 | 21.74 | 18.55 | 19.90 |
| Carbon— | | | | | | | | | |
| Charcoal..... | 12.13 | 4,580 | 3,215 | 2,440 | 1,650 | 14,500 | 15.00 | 13.30 | 14.14 |
| Coke..... | | | | | | | | | |
| Anthracite C'l } | | | | | | | | | |
| Coal— | | | | | | | | | |
| Cumberland.. | 12.06 | 4,900 | 3,360 | 2,550 | 1,730 | 15,370 | 15.90 | 14.28 | 15.06 |
| Coking bituminous..... | 11.73 | 5,140 | 3,520 | 2,680 | 1,810 | 15,837 | 16.00 | 14.45 | 15.10 |
| Cannel..... | 11.80 | 4,850 | 3,330 | 2,540 | 1,720 | 15,080 | 15.60 | 14.01 | 14.76 |
| Lignite..... | 9.30 | 4,600 | 3,210 | 2,490 | 1,670 | 11,745 | 12.15 | 10.78 | 11.46 |
| Peat— | | | | | | | | | |
| Kiln-dried..... | 7.68 | 4,470 | 3,140 | 2,420 | 1,660 | 9,660 | 10.00 | 8.92 | 9.42 |
| Air-dried, 25 per cent water... | 5.76 | 4,000 | 2,820 | 2,240 | 1,556 | 7,000 | 7.25 | 6.41 | 6.78 |
| Wood— | | | | | | | | | |
| Kiln-dried..... | 6.00 | 4,080 | 2,910 | 2,260 | 1,530 | 7,245 | 7.50 | 6.64 | 7.02 |
| Air-dried, 20 per cent water.... | 4.80 | 3,700 | 2,670 | 2,100 | 1,490 | 5,600 | 5.80 | 4.08 | 4.39 |

The above table gives the air required for complete combustion, the temperature attained with different proportions of air, the theoretical value, and the highest practically attainable value under a steam-boiler, assuming that the gases pass off at 320°, the temperature of steam at 75 lbs. pressure, and the incoming air at 60°; also, that with chimney draught twice, and with forced blast only, the theoretical amount of air is required for combustion.

With hickory at \$5 a cord, other woods are worth about as below :

| | |
|-------------------|--------|
| Hickory..... | \$5 00 |
| White oak..... | 4 05 |
| White ash..... | 3 85 |
| Apple..... | 3 50 |
| Red oak..... | 4 45 |
| White beech..... | 3 25 |
| Black walnut..... | 3 25 |
| Black birch..... | 3 15 |
| Hard maple..... | 3 00 |
| White elm..... | 2 90 |
| Red cedar..... | 2 08 |
| Wild cherry..... | 2 75 |
| Soft maple..... | 2 70 |
| Yellow pine..... | 2 70 |
| Chestnut..... | 2 60 |
| Butternut..... | 2 55 |
| White birch..... | 2 40 |
| White pine..... | 2 10 |

Mr. D. K. Clark thus assigns the several portions of the heat of combustion of good coke, as burned in the locomotive :*

| | | |
|--------------------------|-----------------------|---------------------|
| Making steam..... | 10,920 B. T. U. | 73 per cent. |
| Loss at smoke-stack..... | 2,316 " | 16.5 " |
| Ash and waste..... | 764 " | 5.5 " |
| | <hr/> 14,000 B. T. U. | <hr/> 100 per cent. |

and concludes that combustion in the furnace of the locomotive may be, and often is, practically perfect, and anticipates that economy in the formation of steam will only be improved by utilizing heat now wasted at the chimney. The usual maximum evaporation is about 8 times the weight of coke used—a low figure, which is mainly due to the comparatively small proportion of heating-surface adopted. The nearer the composition of the fuel approaches that of coke, the better, as a rule, the economical effect. Coal gives, as an average, about two thirds the effect of coke, as customarily burned ; and its value may be fairly approximated, the composition being known, by assuming the carbon to be the only useful constituent.

* Railway Machinery, p. 122.

Economy in combustion of fuels, where they are used simply in the production of high temperature, is so important a matter, except in those favored localities where the proximity of coal, or of peat-beds, or of forests, renders its waste less objectionable, that the engineer should omit no precaution in the endeavor to secure their perfect utilization.

To secure the greatest economy, it is necessary to adopt a form of grate which, while allowing a sufficient supply of air to pass through it to insure complete combustion, has such narrow air-spaces as to prevent waste of small fragments, by falling through them.

The narrower the grate-bars and the air-spaces, the more readily can losses from this cause and from obstruction of draught be avoided. With a hot fire, however, the difficulties arising from the warping of the bars become so great, that it is only by peculiar devices for interlocking and bracing them that their thickness can be reduced below about $\frac{1}{8}$ of an inch at the top. Many such devices are now in use. In furnaces burning wet fuel, with an ash-pit fire, fire-brick grate-bars are used.

A certain amount of air must usually be allowed to enter the furnace above the grate, to consume those combustible gases which do not obtain the requisite supply of oxygen from below. The carbon, probably, in such cases usually obtains its oxygen from below the grate, while the gaseous constituents of the fuel are consumed by the oxygen coming in above.

Chas. Wye Williams, who made most extended and careful experiments on combustion of fuel, recommended, for ordinary cases, where bituminous coal was burned, a cross area of passage, admitting air above the grate, of one square inch for each 900 pounds of coal burned per hour, or about one square centimetre for each 63 kilogrammes of fuel. This area should be made larger, proportionally, as the thickness of the bed of the fuel is increased, and as the proportion of hydrocarbons becomes greater.

Chilling the gases, before combustion is complete, should be carefully prevented; and comparatively cold surfaces, as

those of a steam-boiler, should not be placed too near the burning fuel. A large combustion-chamber should, where possible, be provided, and more complete combustion may be expected in furnaces of large size, lined with fire-brick, and with arches of the same material, than in a furnace of small size where the fire is surrounded by chilling surfaces, as in a "fire-box steam-boiler."

Finally, the greatest possible amount of heat being developed in combustion, careful provision should be made for completely utilizing that heat.

In a steam-boiler this is accomplished by having large heating-surfaces, and by so arranging the distribution of the adjacent currents of water and of hot gases that their difference of temperature shall be the greatest possible. The gases should enter the flues at that part of the boiler where the temperature is highest, and leave them at the point of lowest temperature. The feed-water should enter as near as possible to the point where the gases pass off to the chimney, and should gradually circulate until evaporation is completed at, as nearly as possible, that part of the boiler nearest to the point of entrance of the heated gases.

Where a small combustion-chamber is unavoidably employed, as in locomotives, various expedients have been devised with the object of producing complete intermixture of gases before entering the tubes. The most common and most successful is a bridge-wall, sometimes depending from the crown sheet, but sometimes rising from the grate, and which, by the production of eddies in the passing current, causes a more thorough commingling of the combustible gases with the accompanying air. None of these devices seem yet to have given such good results as to induce their general adoption.

In the furnaces of steam-boilers it is usually considered advisable to allow the gaseous products of combustion to enter the chimney at a temperature of about 600° Fahr. (315° Cent.),

The management of fires is an important but often neglected branch of instruction in fitting firemen for their special duties. The economy of boiler management is very largely dependent upon the skilful handling of the fuel and the

furnace. In general, the fires should be kept of even thickness, clear of ash and clinkers, and as clean at the sides and in the corners as elsewhere. The depth of the fuel is determined by its nature and size and by the intensity of the draught. Hard coals can be used in greater depth than soft, and large coal in deeper fuel-beds than small. A strong draught demands a thick fire, a mild draught a thin one. With a low chimney and natural draught small anthracite or fine bituminous coal may be most successfully burned in a layer but a hand's breadth in thickness; while with large "steamboat" coal of the hardest varieties and with a heavy forced draught, fires have been actually worked successfully of five times that depth, or more. The secret of success in handling fires is to find the best depth of fire for the conditions existing; to keep that thickness at all times, allowing for the ash that may accumulate; to throw the fuel on the grate at such frequent intervals as will prevent the fire burning into holes or in irregular thickness at different points; to introduce the coal so quickly and with such exactness of direction that no serious loss may occur from the inrush of cold air, and so that every shovelful should go precisely where needed, the place for the next shovelful being at the same instant located. The removal of ash is best done by means of a rake or other tool used under the grate, rather than by stirring and breaking up the bed of fuel by working through the furnace-door. The various forms of shaking grate now in use are often very efficient. For best working, the fire should usually be kept bright beneath, and the ash-pit clear. With light draught, however, and thin fires, it is sometimes advisable, if sufficient steam can be so made, to allow the fire to be less frequently raked out, and some accumulation of ash may be thus produced when working with maximum economy.

"Firing," or "stoking," as the replenishing of the fuel is called, must be done very quickly and skilfully to avoid serious annoyance by variation of steam-pressure and supply. Where several furnaces are in use this difficulty is less likely to be met with, as the fires may be cooled and cleaned in rotation. A skilful man will find it possible to keep steam very steadily with but two furnaces, even.

Ash-pits should not be allowed to become filled with ashes, as the result would be the checking of the draught, the reduction of the steaming capacity of the boiler, and loss of efficiency, even if not the melting down of the grates. It is customary at sea to clean out the ash-pits and send up ashes, throwing them overboard once in every watch of four hours, when in full steaming. If much unburned fuel is found in the ashes, it should be, if possible, cleaned out and returned to the fire, or used elsewhere.

Cleaning fires consists in thoroughly breaking up the mass of fuel on the grate, shaking out all the ashes, quickly raking out all "clinker," as the semi-fused masses of ash and fuel are called, and, after getting a level, clean bed of good fuel, as promptly as possible covering the whole with a layer of fresh coal. This is done, usually, once in four hours at sea and twice a day on land: but different fuels require somewhat different treatment. The work should be performed with the greatest possible thoroughness and dispatch, to avoid serious loss of steam-pressure.

Mr. C. W. Williams' instructions for handling the fires, where bituminous coal is used and an air-supply above the fuel is provided, are substantially as follows:

Charge the furnace from the bridge-end, gradually adding fuel until the dead-plate is reached and the whole grate evenly covered. Never permit the fire to get lower than four or five inches in thickness, of clear and incandescent fuel, uniformly distributed, and laid with especial care along the sides and in the corners. Any tendency to burn into holes must be checked by filling the hollows and securing a level surface. All lumps should be broken until not larger than a man's fist. Clean out the ash-pit so often that there shall be no danger of overheating the grate-bars.

An ash-pit, brightly and uniformly lighted by the fire above, indicates that it is in good order and working well. A dark or irregularly lighted ash-pit is indicative of an uncleaned and badly working fire. The cleaning of the fire is best done, in ordinary working, by a "rake" or other tool working on the

under side of the grates, and not by a "slice-bar" driven into the mass of fuel and above the grate.

Different fuels require different treatment. The principles just stated apply generally, but more, perhaps, to anthracite coals. The soft coals are commonly so disposed on the fire that a charge may have time to coke and its gases to burn before it is spread over the grate; liquid fuels must be so supplied that they may burn completely, at a perfectly uniform rate, and especially in such manner as to be safe from explosive combustion; the same precaution is demanded with the gaseous fuels. Special arrangements of grate and a special routine in working may be, and often are, demanded in such cases.*

The liquid and gaseous fuels are often and successfully burned in conjunction with solid fuels. In such cases the same methods are to be adopted and precautions observed in handling the latter as when burned alone.

The liquid fuels are almost invariably the crude petroleums. They are sometimes burned in a furnace in which they are allowed to drip from shelf to shelf in a series arranged vertically at the front of the furnace, the flame passing to the rear, with the entering current of air supporting their combustion. In many cases they are sprayed into the furnace by a jet of steam which should be superheated and at high pressure. The use of the steam is considered to have a peculiar and beneficial effect, possibly through chemical reactions facilitating the formation of hydrocarbons. The petroleums are all liable to cause accident if carelessly handled, and special precaution must be observed in their application to the production of steam.

The gaseous fuels are seldom used under steam-boilers, except where "natural" gas from gas-wells is obtainable, or where a very large demand or the use of metallurgical processes justifies the construction of gas-generators. Even greater precautions against accidents by explosion are needed than with the liquid fuels. In burning gas, maximum economy is secured by careful apportionment of the air-supply to the gas-consumption, and especially in avoiding excess. The regulation system is not generally economically applicable to boilers.

The stored energy in steam and in water at any pressure

and temperature is easily ascertained by calculation, in accordance with the first law of thermodynamics.

The first attempt to calculate the amount of energy latent in the water contained in steam-boilers, and capable of greater or less utilization in expansion by explosion, was made by Mr. George Biddle Airy,* the Astronomer Royal of Great Britain, in the year 1863, and by the late Professor Rankine† at about the same time.

Approximate empirical expressions are given by the latter for the calculation of the energy and of the ultimate volumes assumed by unit weight of water during expansion, as follows, in British and in metric measures:

$$U = \frac{772(T - 212)^2}{T + 1134.4}; \quad U_m = \frac{423.55(T - 100^2)}{T + 648};$$

$$V = \frac{36.76(T - 212)}{T + 1134.4}; \quad V_m = \frac{2.29(T - 100)}{T + 648}.$$

These formulas give the energy in foot-pounds and kilogrammetres, and the volumes in cubic feet and cubic metres. They may be used for temperatures not found in the tables to be given, but, in view of the completeness of the latter, it will probably be seldom necessary for the engineer to resort to them.

The quantity of work and of energy which may be liberated by the explosion, or utilized by the expansion, of a mass of mingled steam and water has been shown by Rankine and by Clausius, who determined this quantity almost simultaneously, to be easily expressed in terms of the two temperatures between which the expansion takes place.

When a mass of steam, originally dry, but saturated, so expands from an initial absolute temperature, T_1 , to a final absolute temperature, T_2 , if J is the mechanical equivalent of the

* Numerical Expression of the Destructive Energy in the Explosions of Steam Boilers.

† On the Expansive Energy of Heated Water.

unit of heat, and H is the measure, in the same units, of the latent heat per unit of weight of steam, the total quantity of energy exerted against the piston of a non-condensing engine, by unity of weight of the expanding mass, is, as a maximum,

$$U = JT_2 \left(\frac{T_1}{T_2} - 1 - \text{hyp log } \frac{T_1}{T_2} \right) + \frac{T_1 - T_2}{T_1} H.$$

This equation was published by Rankine a generation ago.*

When a mingled mass of steam and water similarly expands, if M represents the weight of the total mass and m is the weight of steam alone, the work done by such expansion will be measured by the expression

$$U = MJT_2 \left(\frac{T_1}{T_2} - 1 - \text{hyp log } \frac{T_1}{T_2} \right) + m \frac{T_1 - T_2}{T_2} H.$$

This equation was published by Clausius in substantially this form.†

It is evident that the latent heat of the quantity m , which is represented by mH , becomes zero when the mass consists solely of water, and that the first term of the second member of the equation measures the amount of energy of heated water which may be set free, or converted into mechanical energy by explosion. The available energy of heated water, when explosion occurs, is thus easily measurable.

36. The Specific Heats of Water and Steam vary somewhat with temperature; this variation is noted with all solids, and occurs with the vapors, although in vastly less degree; and this is one point in which they are distinguished from the gases. For all the purposes of the engineer the specific heat of either saturated steam or of steam-gas may be taken at the value obtained by Regnault, 0.305, the quantity of heat, in thermal units, demanded to raise the temperature of units of weight of saturated steam one degree, while still keeping it saturated by

* Steam-engine and Prime Movers, p. 387.

† Mechanical Theory of Heat, Browne's translation, p. 283.

the evaporation of additional water; which latter process demands the transformation of 0.695 unit. of heat.

The specific heat of isolated steam-gas, or superheated steam, is given by Regnault as 0.48051, and constant.

The specific heat of water was determined by Regnault * very carefully and exactly, and the figures so obtained have been found capable of being very accurately represented by the following empirical formula of Rankine:†

$$C = 4 + 0.000000309(t - 39^\circ.1)^2,$$

in which t is the temperature on the common Fahrenheit scale.

The total heat demanded from t_1 to t_2 would thus be

$$h = \int_{t_1}^{t_2} C dt = t_2 - t_1 + 0.000000103[(t_2 - 39^\circ.1)^3 - (t_1 - 39^\circ.1)^3];$$

and the mean specific heat for such a range of temperature is

$$\frac{h}{t_2 - t_1} = 1 + 0.000000103[(t_2 - 39^\circ.1)^2 + (t_2 - 39^\circ.1)(t_1 - 39^\circ.1) + (t_1 - 39^\circ.1)^2].$$

On the Centigrade scale these expressions become

$$C = 1 + 0.000001(t - 4^\circ)^2,$$

$$h = t_2 - t_1 + 0.00000033[(t_2 - 4^\circ)^3 - (t_1 - 4^\circ)^3],$$

$$\frac{h}{t_2 - t_1} = 1 + 0.00000033[(t_2 - 4^\circ)^2 + (t_2 - 4^\circ)(t_1 - 4^\circ) + (t_1 - 4^\circ)^2].$$

The specific heat of *ice* is given by M. Person as 0.504.

Regnault's and Wiedemann's experiments, made on simple gases, and on carbonic oxide, which is formed without condensation, proved that in these cases the specific heat between 0° and 200° C. is constant; whilst their experiments on gases

* Mem. of the Academy of Sciences, 1847.

† Trans. Royal Soc. of Edinburgh, 1851; Steam-engine, p. 246.

formed with condensation show that the specific heat varies, the mean being given in the following empirical formulæ:

$$\begin{aligned} \text{For CO}_2 = 44 \text{ gr. C.} &= 8.41 + 0.0053t \quad \left\{ \begin{array}{l} \text{Mean of Regnault} \\ \text{and Wiedemann.} \end{array} \right. \\ \text{" NO} = 44 \text{ " } &= 8.96 + 0.0028t \\ \text{" C}_2\text{S}_4 = 76 \text{ " } &= 10.62 + 0.007t, \text{ Regnault.} \\ \text{" NH}_3 = 17 \text{ " } &= 8.51 + 0.00265t, \text{ Wiedemann.} \\ \text{" C}_4\text{H}_4 = 28 \text{ " } &= 9.42 + 0.0115t, \text{ Wiedemann.} \end{aligned}$$

37. The Computation of Latent and Total Heat of Steam is readily made by means of formulas given by Regnault or based upon his work, which covered a wide range of temperature—from a little below the freezing-point to about $375^\circ \text{ F. (} 190^\circ \text{ C.)}$. The following is the formula of Regnault for latent heat as slightly modified and corrected by Rankine for the British and metric systems, respectively:

$$l = 1091.7 - 0.695(t - 32^\circ) - 0.000000103(t - 39^\circ.1)^2;$$

$$l_m = 606.5 - 0.695t - 0.000000333(t - 4^\circ)^2;$$

or, approximately, as given by the investigator,

$$l = 1092 - 0.7(t - 32^\circ);$$

$$= 968 - 0.7(t - 212);$$

$$= 1147 - 0.7t;$$

$$l_m = 606 - 0.7t.$$

The *total heat of evaporation* is the sum of the latent and sensible heats, and may be taken as

$$h = C(t_1 - t_2) + l_1;$$

$$= 1091.7 + 0.305(t - 32^\circ);$$

$$h_m = 606.5 + 0.305t;$$

in which the "total heat" measured is that *from* t_2 *at* t_1 , the original temperature of the water and that of evaporation, and the formulas given being based on the assumption that t_1 is

taken at the melting-point of ice. For any other temperature the following will give satisfactorily exact measures:

$$\begin{aligned} h &= 1092 + 0.3(t_1 - 32) - (t_2 - 32^\circ); \\ &= 1146 + 0.3(t_1 - 212) - (t_2 - 32^\circ); \\ h_m &= 606.5 + 0.3t_1 - t_2; \end{aligned}$$

h being obtained in British measures and h_m in metric.

For *steam-gas*,

$$h = 1092 + 0.48(t_2 - t_1).$$

Professor Unwin proposes the following for the latent heat of vaporization of steam:

$$l_m = 799 - \frac{894}{(7.5030 - \log p)^{0.8}},$$

which is found to be extremely exact. He also obtains for the expansion during change of state,

$$\Delta v_m = 10.821 \frac{l_m}{p(7.5030 - \log p)};$$

p being given in millimetres of mercury.

38. Factors of Evaporation measure the relative amount of heat demanded to effect the heating of water from a given temperature, t_1 , and its vaporization at a higher temperature, t_2 , and to simply produce vaporization at the boiling-point under atmospheric pressure, which latter is now usually taken as a standard. The value of this factor of evaporation is evidently

$$f = 1 + \frac{0.3(t_1 - 212^\circ) + (212^\circ - t_2)}{966.1}, \text{ nearly.}$$

In Table XII, Appendix, are values of such factors, calculated as above by Mr. Kent.

It is seen that the relative cost of using feed-water at any one temperature as compared with the use of water at any other temperature is as the reciprocal of their factors of evapo-

rization. Thus, if feed-water can be supplied, by means of a heater, at 210° F., where previously drawn from the mains at 50° , the relative cost of making steam will be, at 100 pounds pressure, by gauge, $\frac{1043}{1208} = 0.86$, and a gain of fourteen per cent will be effected. As will be seen later, these tables are very useful in reducing the data obtained in trials of steam-boilers to the standard.

39. Regnault's Tables have been reproduced in many forms, usually with additions. The Appendix, among other tables, contains the data obtained by Regnault, and these values are accepted as standard universally. The table there given exhibits the temperatures and corresponding pressures of saturated steam throughout the full range now used in steam-boilers and far beyond; the quantity of heat, sensible and latent, in unity of weight; the total heat of evaporation and the density of the steam. Reference to these tables is vastly more convenient than calculation. Should it be necessary, or desirable, however, to make such calculations, the formulas already given will furnish the means. They also permit the calculation of data beyond the limits of Regnault's experiments, and are probably practically correct far beyond any pressure likely to become familiar in the operation of steam-boilers. Regnault's limit was at 230° C. (446° F.). Rankine's formula has been used beyond it.

The formulas used in these calculations are also given in the Appendix, grouped for convenience of reference. The Appendix also contains all the numerical constants ordinarily needed in computation of the quantities demanded in determining the efficiencies and performance of engines or boilers or both.

CHAPTER III.

RESULTS OF STANDARD METHODS; APPARATUS.

40. The Results of Trials actually conducted under acceptable conditions, and with all the precautions which have been suggested, are illustrated by the following examples:

The first case is a trial which was carried out in accordance with the above programme. The measurements of the feed-water were made by passing the water through a Worthington metre into two wooden tanks located on Fairbanks' Standard Platform Scales. The pipe connections were so arranged that one tank could be filled and weighed while the other tank was being emptied into the boiler.

Each tank was filled once every half hour. As soon as the tank was full and the pumping into the boiler commenced, the temperature of the feed-water was taken by sensitive thermometers reading to one-tenth of a degree.

All water-measurements, as in all instances of careful work, were made by weight rather than by volume, and systems of checking were devised and practised whenever practicable. The apparatus was all carefully standardized and repeatedly re-examined and tested as opportunity offered.

The measurements of the coal were effected by weighing the coal previous to its being wheeled into a pile in the coal-room. The second weighing was made when the coal was fed into the furnace. As far as it was possible, the furnace was supplied with coal at intervals of every half hour, so as to correspond as nearly as could be to the feeding of the water.

After the completion of the test, a careful analysis of the coal was made, to determine upon a sufficiently large scale its calorific power and the quantity of contained moisture. The

steam from the boiler was condensed by means of a continuously acting calorimeter, formed by placing four tanks on Fairbanks Standard Platform Scales. The steam from the boiler was passed through a surface-condenser having a condensing surface of 631 sq. ft. As fast as the steam was condensed from the boiler it was received in small tanks located on platform-scales. These tanks were similar in size to the feed-water tanks, and were so arranged as to be filled and emptied once every half hour, one tank receiving the condensed water from the boiler while the other was being emptied.

The condenser was supplied with a large volume of cold water from a weir just outside of the works, and after flowing through the condenser and thereby cooling the steam and receiving therefrom the contained heat, this water was caught in two large tanks placed on platform-scales. These tanks were also arranged so that one tank could be emptied while the other was being filled, and were of sufficient capacity so as to insure catching all of the water required for half an hour's run in the condenser. The temperature of the inlet water of the condenser, of the outlet water, and of the condensed steam were carefully noted by means of thermometers reading to a tenth of a degree. Readings of the inlet water and of the condensed steam were taken once every half hour at the same time that the quantities of the water in the tanks were weighed. Inasmuch as the outlet to the condenser varied considerably in temperature, readings on this were taken every five minutes during the entire time of the test. It will thus be seen that a very correct average of the amount of heat given to the condenser was obtained. The quantity of air supplied by the blowers to the furnace was measured by continuously acting anemometers placed in the supply-pipes. The readings of the anemometers were checked by means of the number of revolutions of the blowers and their cubic feet per revolution.

The steam-pressure was kept by a recording pressure-gauge, which was checked by an exceedingly delicate and sensitive gauge, which previously, and subsequently to the test,

was carefully verified by means of a mercury column. Constant records of the hygrometer, barometer, and thermometers, both in the boiler-room and of the external air, were kept during the entire period of the test.

It will be seen from the above, that all of the processes and measurements were kept in duplicate in such a way as to afford a constant check on each other and preclude the possibility of any errors.

Samples of steam were taken in a small calorimeter for the purpose of ascertaining whether the boiler supplied wet steam.

The following is a brief condensed summary:

EFFICIENCY AS PER TEST, 7.50 A.M. to 7.50 A.M.

| | | | |
|-----------------------------------------------------------|------------|-------------|----------------|
| Total heat of boiler | 64,536,613 | heat-units. | |
| Steam..... | 42,933,141 | " " | 66.6 per cent. |
| Heat escaping in flue-gases.... | 9,669,036 | " " | 15 " " |
| Radiated heat..... | 5,162,939 | " " | 8 " " |
| Heat to vaporize moisture in coal | 141,372 | " " | 0.2 " " |
| Heat to vaporize moisture in air supplied to furnace..... | 345,978 | " " | 0.4 " " |
| Leakage..... | 3,531,645 | " " | 4.0 " " |
| " from pump..... | 127,936 | " " | 0.2 " " |
| Heat absorbed by fire-brick.... | 2,581,645 | " " | 4.0 " " |
| Unaccounted for..... | 1,092,941 | " " | 1.6 " " |

In the trial of an upright boiler reported on by Sir Frederick Bramwell, in 1876, coke being used as the fuel and wood in starting the fires, the following data* were obtained:

| | | |
|----------------------------------------|------------------|----------|
| Ash and moisture | 43.79 | lbs. |
| Combustible..... | 194.46 | " |
| Total fuel..... | 238.25 | " |
| Air used per pound combustible..... | 17 $\frac{1}{8}$ | " |
| Heat generated, net..... | 2,798,312 | B. T. U. |
| " " per lb. fuel..... | 11,745 | " " |
| " " available, net..... | 2,101,700 | " " |
| Water evaporated. | 1,620 | lbs. |
| The efficiency of the furnace was..... | 0.643 | " |

* Conversion of Heat into Work. Anderson.

The balance-sheet stands thus:

Dr.

Available heat.....2,101,700 B. T. U.

Cr.

Per Cent.

| | | | |
|-------|---------------------------------------|-----------|----------|
| 88.29 | Heat expended in evaporation..... | 1,855,900 | B. T. U. |
| 7.03 | Displacing atmosphere..... | 147,720 | " " |
| 3.35 | Loss by conduction and radiation..... | 70,430 | " " |
| .05 | Heat in ashes..... | 1,129 | " " |
| 1.26 | Unaccounted for..... | 25,521 | " " |

100.00

2,101,700

In trials conducted by the Author, for a committee of the American Institute, of which he was chairman, in testing a number of different types of boiler,* a surface-condenser was employed to condense *all* steam made, and results thus for the first time obtained which gave exact measures of net efficiency, the quality of all steam made being determined.

In calculating the results from the record of the logs, the committee first determined the amount of heat carried away by the condensing water by deducting the temperature at which it entered from that at which it passed off. To this quantity is added the heat which was carried away by evaporation from the surface of the tank, as determined by placing a cup of water in the tank at the top of the condenser at such height that the level of the water inside and outside the cup were the same, noting the difference of temperatures of the water in the cup and at the overflow, and the loss by evaporation from the cup. The amount of evaporation from the surface of the water in the cup and in the condenser, which latter was exposed to the air, was considered as approximately proportional to the tension of vapor due their temperatures, and was so taken in the estimate. The excess of heat in the water of condensation over that in the feed-water also evidently came from the fuel, and this quantity was also added to those already mentioned.

* See Transactions, 1871; also, Report on Mechanical Engineering at Vienna International Exhibition, 1873, R. H. T.

The total quantities were, in thermal units, as follows :

| | |
|--------|---------------|
| A..... | 34,072,058.09 |
| B..... | 48,241,833.60 |
| C..... | 24,004,601.14 |
| D..... | 38,737,217.57 |
| E..... | 11,951,002.10 |

These quantities, being divided by the weight of combustible used in each boiler during the test, will give a measure of their relative economical efficiency; and, divided by the number of square feet of heating-surface, will indicate their relative capacity for making steam. But as it was the intention of the committee to endeavor to establish a practically correct measure that should serve as a standard of comparison in subsequent trials, it was advisable to correct these amounts by ascertaining how and where errors have entered, and introducing the proper correction. There were two sources of error that are considered to have affected the result as above obtained. The tank being of wood, a considerable quantity of water entered it, leaked out again at the bottom, without increase of temperature, instead of passing through the tank and carrying away the heat, as it is assumed to have done in the above calculation. The meters also registered rather more water than actually passed through them, and this excess assists in making the above figures too high. The sum of these errors the committee estimated at 4 per cent of the total quantity of heat carried away by the condensing water. The other two quantities were considered very nearly correct.

Making these deductions, we have the following as the total heat, in British thermal units, which was thrown into the condenser by each boiler :

| | |
|--------|---------------|
| A..... | 32,751,835.34 |
| B..... | 46,387,827.10 |
| C..... | 23,066,685.39 |
| D..... | 37,228,739.07 |
| E..... | 11,485,777.35 |

That the figures thus obtained are very accurate, is shown by calculating the heat transferred to the condenser by the Root and the Allen boilers (both of which superheated their

steam), by basing the calculation on the temperature of the steam in the boiler, as given by the thermometer, the results thus obtained being 32,723,681.76 and 46,483,322.5, respectively.

Dividing these totals by the pounds of combustible consumed by each boiler, we get as the quantity of heat per pound, and as a measure of the relative economic efficiency:

| | |
|--------|-----------|
| A..... | 10,281.53 |
| B..... | 10,246.92 |
| C..... | 10,143.66 |
| D..... | 10,048.24 |
| E..... | 10,964.94 |

Determining the weight, in pounds, of water evaporated per square foot of heating-surface per hour, we get as a measure of the steaming capacity:

| | |
|--------|------|
| A..... | 2.65 |
| B..... | 3.59 |
| C..... | 2.83 |
| D..... | 3.10 |
| E..... | 1.92 |

The quantity of heat per pound of combustible, as above determined, being divided by the latent heat of steam at 212° Fahrenheit (966°.6), gives as the equivalent evaporation of water at the pressure of the atmosphere, and with the feed at a temperature of 212° Fahrenheit:

| | |
|--------|-------|
| A..... | 10.64 |
| B..... | 10.60 |
| C..... | 10.49 |
| D..... | 10.40 |
| E..... | 10.34 |

For general purposes this is the most useful method of comparison for economy.

The above figures afford a means of comparison of the boilers, irrespective of the condition (wet or dry) of the steam furnished by them. All other things being equal, however, the committee consider that boiler to excel which furnishes the driest steam; provided that the superheating, if any, does not exceed about 100°.

In this trial the superheating was as follows:

| | |
|--------|--------|
| A..... | 16°.08 |
| B..... | 13°.23 |
| C..... | 0. |
| D..... | 0. |
| E..... | 0. |

As the boilers C, D, E did not superheat, it became an interesting and important problem to determine the quantity of water carried over by each with the steam. This we are able, by the method adopted, to determine with great facility and accuracy.

Each pound of saturated steam transferred to the condensing water the quantity of heat which had been required to raise it from the temperature of the water of condensation to that due to the pressure at which it left the boiler, *plus* the heat required to evaporate it at that temperature. Each pound of water gives up only the quantity of heat required to raise it from the temperature of the water of condensation to that of the steam with which it is mingled. The total amount of heat is made up of two quantities, therefore, and a very simple algebraic equation may be constructed which shall express the conditions of the problem:

Let

H = heat-units transferred per pound of steam.

h = heat-units transferred per pound of water.

U = total quantity of heat transferred to condenser.

W = total weight of steam and water, or of feed-water.

x = total weight of steam.

$W - x$ = total weight of water primed.

Then

$$Hx + h(W - x) = U; \text{ or } x = \frac{\frac{U}{h} - W}{\frac{H}{h} - 1}.$$

Substituting the proper values in this equation, we deter-

mine the absolute weights and percentages of steam and water delivered by the several boilers as follows:

| | Weight of Steam. | Weight of Water. | Percentage of Water Primed to Water Evaporated. |
|--------|------------------|------------------|-------------------------------------------------------|
| A..... | 27,896. | 0. | 0. |
| B..... | 39,670. | 0. | 0. |
| C..... | 19,782.94 | 645.06 | 3.26 |
| D..... | 31,663.35 | 2,336.65 | 6.9 |
| E..... | 9,855.6 | 296.9 | 3. |

And the amount of water, in pounds, actually evaporated per pound of combustible:

| | |
|--------|------|
| A..... | 8.76 |
| B..... | 8.76 |
| C..... | 8.70 |
| D..... | 8.55 |
| E..... | 9.41 |

Comparing the above results, the committee were enabled to state the following order of capacity and of economy in the boilers exhibited, and their relative percentage of useful effect, as compared with the economical value of a steam-boiler that should utilize all of the heat contained in the fuel:

| | Steaming Capacity. | Economy of Fuel. | Percentage of Economical Effect. |
|--------|--------------------|------------------|----------------------------------------|
| A..... | No. 4 | No. 2 | 0.709 |
| B..... | No. 1 | No. 3 | 0.707 |
| C..... | No. 3 | No. 4 | 0.699 |
| D..... | No. 2 | No. 5 | 0.693 |
| E..... | No. 5 | No. 1 | 0.756 |

The results obtained as above, and other very useful determinations derived from this extremely interesting trial, were given in the table, as a valuable standard set of data with which to compare the results of future trials, and as a useful aid in judging of the accuracy of statements made by boiler-venders in the endeavor to effect sales by presenting extravagant claims of economy in fuel.

Mr. Druitt Halpin found the following net results of test of a variety of English-built boilers:

| No. | DESCRIPTION OF BOILER. | POUNDS WATER EVAPORATED. | | THERMAL UNITS. | | | Efficiency. | G figure of merit = units, per sq. ft. per hour × efficiency. |
|-----|------------------------|----------------------------------------------|--------------------------------------------|----------------|------------------------------------------------------------|-----------------|-------------|------------------------------------------------------------------------|
| | | Per square foot of heating-surface per hour. | Per pound of fuel from and at 212 degrees. | In fuel. | Transmitted per hour per sq. ft. heating-surface per hour. | Per pound fuel. | | |
| 1 | Field | 4.57 | 8.83 | | 4,414 | 8,529 | .. | |
| 2 | Field | 2.28 | 10.83 | | 2,202 | 10,461 | .. | |
| 3 | Field | 2.57 | 10.93 | | 2,482 | 10,558 | .. | |
| 4 | Portable { | 1.52 | 10.23 | 14,718 | 1,468 | 9,882 | 67 | 98,356 |
| 5 | Portable { | 2.26 | 10.49 | 14,718 | 2,183 | 10,133 | 68 | 148,444 |
| 6 | Portable { | 1.76 | 11.81 | 14,718 | 1,700 | 11,408 | 77 | 130,900 |
| 7 | Portable { | 3.56 | 9.93 | | 3,438 | 9,592 | .. | 118,248 |
| 8 | Lancashire | 1.57 | 12.83 | 15,715 | 1,516 | 12,393 | 77 | 108,248 |
| 9 | Lancashire | 2.83 | 9.89 | 13,833 | 2,733 | 9,553 | 68 | 185,844 |
| 10 | Lancashire | 1.88 | 12.25 | 15,715 | 1,816 | 11,833 | 75 | 136,200 |
| 11 | Jacketed | 4.70 | 7.7 | 14,805 | 4,505 | 7,500 | 50 | 229,750 |
| 12 | Lancashire | 2.57 | 10.9 | 15,715 | 2,482 | 10,529 | 67 | 166,294 |
| 13 | Compound | 1.43 | 11.51 | 14,296 | 1,381 | 11,125 | 78 | 107,718 |
| 14 | Loco. (Webb) | 9.83 | 10.28 | 14,004 | 9,495 | 9,930 | 70 | 664,650 |
| 15 | Loco. (Marie) | 4.62 | 10.65 | 14,600 | 4,462 | 10,287 | 70 | 312,340 |
| 16 | Loco. { | 12.57 | 8.22 | 13,550 | 12,142 | 7,940 | 58 | 704,236 |
| 17 | Loco. { | 13.73 | 8.94 | 13,550 | 13,263 | 8,636 | 63 | 835,569 |
| 18 | Loco. { | 6.76 | 10.01 | 13,550 | 6,530 | 9,669 | 71 | 463,630 |
| 19 | Loco. { | 7.39 | 11.2 | 13,550 | 7,138 | 10,819 | 77 | 549,626 |
| 20 | Torpedo | 12.54 | 8.37 | 14,727 | 12,113 | 8,085 | 54 | 654,102 |
| 21 | Torpedo | 14.86 | 7.78 | 14,727 | 14,354 | 7,523 | 51 | 732,054 |
| 22 | Torpedo | 17.90 | 7.49 | 14,727 | 17,291 | 7,235 | 49 | 847,259 |
| 23 | Torpedo | 20.74 | 7.04 | 14,727 | 20,034 | 6,800 | 46 | 921,564 |
| 24 | | a | b | c | d | e | f | g |

The "locomotive" boiler is found to be more efficient as a part of the engine and on the track than when mounted as a stationary boiler, an unexpected result.

41. The Quality of Steam made in any boiler, or as supplied to an engine, is hardly less important than the quantity. When the steam is required for heating purposes simply, or even when all the heat issuing as waste, necessary or other, from the exhaust-ports of a non-condensing engine cylinder can be utilized for useful and paying purposes, this is a matter of no importance; but when it is essential that loss in the engine shall be made a minimum, and that the engine shall have maximum efficiency, the quality of the steam becomes exceedingly important. Dry steam is very much more efficient as a working substance in the steam-engine than wet; since, where the latter is supplied from the boiler, the waste by cylinder-condensation is greatly increased—and so greatly that the more obvious direct loss by the passing of heat through the engine in unavailable form, hot water acting as its vehicle, becomes comparatively small. The determination of the quality

of steam by any boiler is thus as important as the measure of its apparent evaporation.

The difference between the apparent and the actual evaporation is often very great. A good boiler properly managed will usually "prime" less than five per cent, even though having no superheating-surface, and less than two per cent may usually be hoped for. Steam is often made practically dry. But a hard-worked boiler, or one having defective circulation, will often prime ten or twenty per cent; and cases have been found in the experience of the Author in which the quantity of water carried out of the boiler by the current of steam exceeded the weight of the steam itself. It has thus happened that, where no measure of this defect has been made, the apparent evaporation only being reported, the quantity of water said to have been evaporated has equalled, and sometimes has even greatly exceeded, the theoretically possible evaporation of an absolutely perfect boiler. It is thus essential that, when the apparent evaporation has been determined by trial, the quantity of water entrained with the steam be measured and deducted, and then *real* evaporation thus ascertained and reduced for the standard conditions. Under ordinarily good conditions, a real evaporation of ten or eleven times the weight of the fuel, corresponding to an efficiency of 0.75 to 0.80, represents the best practice, and a real evaporation of twelve of water by one of combustible, from and at the boiling-point, or an efficiency of eighty per cent, is rarely observed under the usually best conditions of steam-boiler practice. Where more than the efficiency here given as probable is reported, the work should be very carefully revised, and errors sought until absolute certainty is secured.

Trials not including calorimetric measurement of the water entrained with the steam are comparatively valueless, and should be rejected in any important case. Reports of extraordinary economy are often based on this kind of error. The experiments of M. Hirn at Mulhouse showed an average of about 5 per cent priming; Zeuner makes it approximately from $7\frac{1}{2}$ to 15 per cent; while the experiments of the Author at the American Institute in 1871 give from 3 to 6.9 per cent.

A recently devised method of measuring the amount of moisture in the steam is to introduce into the boiler with the feed-water sulphate of soda, and at intervals to draw from the lower gauge-cock a small amount of water, and also from the steam, condensing either by a coil of pipe in water or a small pipe in air. A chemical analysis gives the proportion of sulphate of soda in each portion, and the quotient of the proportion of sulphate of soda in the portion from the steam by the proportion in that from the water gives the ratio of water entrained, as steam does not carry sulphate of soda, which is only brought over by the hot water entrained. This method was used by Professor Stahlschmidt at the Düsseldorf Exhibition Trials.

42. The Calorimeters used in determining the quantity of moisture in steam have several forms, widely differing in construction, and to some extent in value. They nearly all embody the same principles, however. The objects sought to be attained in their construction are: The exact measurement of the weight of steam received by them from the boiler, and of its temperature and pressure at the boiler; the determination of the weight of water used in its condensation and the range of temperature through which it is raised in the operation; the reduction of wastes of heat in the calorimeter to a minimum, and the exact measurement of that waste if it is sensibly or practically noticeable.

The Barrel or Tank Calorimeter as employed by the Author, is the simplest form of this instrument which has been produced. It consists of a strong barrel or tank, of hard wood, absorbing little of either water or heat, and having a movable cover. This tank is mounted on platform-scales capable of accurate adjustment and having as fine readings as possible. It is filled with water to within about one fourth its height from the top, and the steam is led into it through a rubber tube or hose of sufficient capacity to supply the steam to the amount of one eighth or one tenth the weight of the water in three or five minutes. A steam-gauge of known accuracy gives the boiler-pressure, and the corresponding temperature and total heat of the steam are ascertained from the steam-tables.

In using this apparatus the steam is rapidly passed into the mass of water contained in the tank, until the scales show that the desired quantity has been added. The steam is so directed by varying the position of the end of the tube, and by inserting it so deeply in the water that the whole mass is very thoroughly stirred, and a very perfect mixture secured of condensing water with the water of condensation; and so that the temperatures indicated by the inserted thermometer shall be the real mean temperature of the mass. The weights and

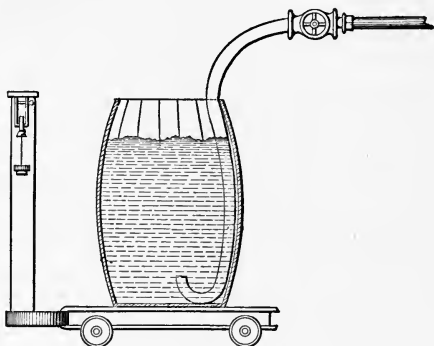


FIG. 1.—THE CALORIMETER.

temperatures are then inserted in the log of the trial, as below, and the proportion of water brought over with the steam is thence easily calculable. The thermometers employed usually read to tenths of a degree Fahrenheit, or to twentieths of a centigrade degree, accordingly as the one or the other scale is employed. Readings must be made with the greatest possible accuracy, and in sufficient number to insure a satisfactorily exact mean. With good thermometers and scales, a reliable gauge, and care in operation, good results can be obtained by averaging a series of trials.*

The *Hirn Calorimeter* is substantially the same as the above, with the addition of an apparatus for stirring the water

* Report on Boiler Trial, Trans. A. S. M. E. 1884, vol. vi.

in the tank to insure thorough mixture and readings of temperature of condensing water exactly representative of the true mean temperature of the mass after the introduction of the steam. This is not an essential feature of the apparatus, if the Author may judge by his own experience, provided the jet of entering steam is so directed as to cause rapid circulation. No stirring apparatus could operate more efficiently than the force of the steam itself, properly directed. Hirn was probably the first (1868) to attempt the determination of the quality of steam as delivered from steam-boilers.* A similar apparatus was used at the trials of the Centennial International Exhibition, Philadelphia, 1876.†

43. The Theory of the Calorimeter is as follows:‡

Each pound of saturated steam transferred to the condensing water the quantity of heat which had been required to raise it from the temperature of the water of condensation to that due to the pressure at which it left the boiler, *plus* the heat required to evaporate it at that temperature. Each pound of water gives up only the quantity of heat required to raise it from the temperature of the water of condensation to that of the steam, with which it is mingled. The total amount of heat is made up of two quantities; therefore, and a very simple algebraic equation may be constructed, which shall express the conditions of the problem:

Let, as in § 258,

H = heat-units transferred per pound of steam;

h = heat-units transferred per pound of water;

U = total quantity of heat transferred to condenser;

W = total weight of steam and water, or of feed-water;

x = total weight of steam;

$W - x$ = total weight of water primed.

* Bulletin de la Société Industrielle de Mulhouse, 1868-9.

† Reports of Judges, vol. vi.

‡ First published by the Author, who had not then become aware of the work done by M. Hirn, in Trans. Am. Inst. Report on Boiler Trial, 1871. See also Vienna Reports, vol. iii. p. 123.

Then

$$Hx + h(W - x) = U; \text{ or } x = \frac{\frac{U}{h} - W}{\frac{H}{h} - 1} = \frac{U - Wh}{H - h}.$$

Substituting the proper values in this equation, we determine the absolute weights and percentages of steam and water delivered by the boiler.

Or, let

Q = quality of the steam, dry saturated steam being unity ;

H' = total heat of steam at observed pressure ;

T = " " " water " "

h' = " " " condensing water, original ;

h_1 = " " " " " final.

And we have the equivalent expression, as written by Mr. Kent,

$$Q = \frac{1}{H' - T} \left[\left(\frac{W'}{w} (h' - h_1) \right) - (T - h_1) \right].$$

The value of the quantity U is obtained by multiplying the weight of water in the calorimeter originally by the range of temperature caused by the introduction of the steam from the boiler. Mr. Emery employs another form, as below, in which Q is the quality of steam as before ; W the weight of condensing water ; w the weight added from the boiler ; T the temperature due the steam-pressure in the boiler ; t the initial and t_1 the final temperature of the calorimeter ; l the latent heat of evaporation of the boiler-steam ; and x the weight of steam corresponding to l . Thus,

$$x = \frac{W(t_1 - t) - w(T - t_1)}{l},$$

and

$$Q = \frac{x}{w} = \frac{W(t_1 - t) - w(T - t_1)}{lw}.$$

The following expressions give the quality of steam as computed from metric data supplied by the calorimeter, and Bosscha's corrections being introduced for specific heat of water at varying temperatures: *

w_1 = weight of condensing water ;

p_1 = absolute pressure of steam ;

t_1 = the initial temperature of condensing water ;

t_2 = the final temperature of the water ;

t_3 = the temperature of steam in boiler ;

w_2 = cold-water weight ;

w_3 = weight of steam condensed ;

x' = percentage of steam in the mixture from the boiler, uncorrected for specific heat of water ;

x = same as x' , but with Bosscha's correction ;

$$x' = \frac{(t_2 - t_1)w_2 - (t_3 - t_2)w_3}{(606.5 - 0.695t_3)w_3},$$

$$x = \frac{(t_2 - t_1)w_2 - (t_3 - t_2)w_3 - 0.00011[(t_3^2 - t_1^2)w_2 - (t_3^2 - t_2^2)w_3]}{(606.5 - 0.695t_3)w_3}.$$

Mr. Nystrom has employed the Hirn calorimeter, substituting *ice* for cold water as a condensing medium.† In this case, adopting his notation,

w = pounds of cold water put into the barrel ;

h = units of heat per pound of w when cold and above 32° ;

I = pounds of ice put into the barrel ;

W = pounds of heated water in the barrel after the completion of the experiment ; that is, including the weight of the condensed steam ;

h' = units of heat per pound of W above 32° ;

f = pounds of foam or water carried over with the steam into the barrel ;

S = pounds of saturated steam blown into the barrel ;

* Proc. Brit. Inst. C. E., 1888, No. 2306.

† Pocket-book; Humidity of Steam, p. 572.

H = units of heat per pound of the steam S ;

H' = units of heat per pound of the foam f ;

p = pounds of steam and foam carried over from the boiler into the barrel;

P = units of heat passed over with the steam and foam into the barrel;

The weight p must then be equal to the sum of the weights of the steam S and foam f , which is evidently the same as the difference between the weights W and w .

That is,

$$p = S + f = W - w.$$

The total units of heat P passed over with the steam S and foam f must then be :

$$P = HS + H'f = Wh' - wh.$$

By solving this formula for the steam S , we have :

$$S = \frac{P - H'f}{H} = W - w - f.$$

$$S = p - f;$$

$$H(p - f) = P - H'f;$$

$$Hp - Hf = P - H'f;$$

$$f(H - H') = Hp - P.$$

From this formula we have the weight of foam carried over with the steam from the boiler into the barrel ; namely,

$$f = \frac{Hp - P}{H - H'}.$$

But $P = Wh' - wh$, which, inserted in the formula, gives :

$$\text{Pounds of foam, } f = \frac{Hp + wh - Wh'}{H - H'}.$$

The percentage of humidity of the steam will then be :

$$\% = \frac{100f}{p}.$$

The Formula (8) is ready for use of the data obtained by the calorimeter when $p = W - w$, and when no ice is used.

For the melting of 1 pound of ice requires 142.65 units of heat, according to Regnault's delicate experiments, but for the caloric experiments on humidity of steam 142.6 units of steam will be more correct. Then the units of heat required to melt the ice to water of 32° by the steam in the barrel will be $142.6I$, and the heat required to raise the temperature of that water from 32° to that of W when the experiment is completed will be Ih' . But the weight of ice melted to water is included in the weight W ; the heat passed with the steam from the boiler into the ice and cold water will be,

$$P = Wh' - wh + 142.6I.$$

This formula, inserted for P in Formula (7), will give the weight of foam passed with the steam from the boiler into the ice and cold water.

$$f = \frac{Hp + wh - (Wh' + 142.6I)}{H - H'}.$$

When no cold water is used, but the humid steam is blown into only ice in the barrel, then the weight of foam will be :

$$f = \frac{Hp - (Wh' + 142.6I)}{H - H'}.$$

The percentage of humidity will be in either case

$$\% = \frac{100f}{p}.$$

For humid steam, $Hp > P$.

For saturated steam, $Hp = P$.

For superheated steam, $Hp < P$.

When the steam is superheated, the formulas give a negative value of f .

The following data and results are given in illustration of this method by its author :

Example.—Steam-pressure by gauge, 98 pounds.

$$H = 1184.6 \text{ and } H' = 308.7.$$

Weight of empty barrel with top, 64.25 lbs.

$I = 80.5$ pounds of ice, 144.75 "

$w = 287.25$ pounds of water at $71^{\circ} h = 39.015$, 432.00 "

$W = 404$ pounds of mixture at $136^{\circ} h = 104.2$, 468.25 "

$p = 36.25$ pounds of steam and foam.

Formula 11.

$$f = \frac{1184.6 \times 36.25 + 287.25 \times 39.015 - (404 \times 104.2 + 142.6 \times 80.5)}{1184.6 - 308.7} = 0.653 \text{ lbs.}$$

$$\text{Humidity of the steam, } \% = \frac{100 \times 0.653}{36.25} = 1.8 \text{ per cent.}$$

In the use of the calorimeter it may be reasonably expected that errors may be made, by careful work, something less than one per cent. The results of good work should agree within less than one-half per cent. Thermometers should read within one-tenth of a degree; steam-gauges should be correct within less than one or two pounds, and weights within one per cent or less. The latter are the most trying quantities to measure with satisfactory accuracy. Generally the lower the initial temperature of the calorimeter the better the results, and it is often thought advisable to cool the condensing water, by the use of ice, down to the melting point. The amount of steam introduced should be as great as is possible without causing so high a final temperature as to cause the production of troublesome quantities of vapor in the calorimeter.

If Q exceeds unity, the steam is superheated by the amount

$$\frac{(Q-1)l}{0.48} = 2.0833l(Q-1);*$$

and if less than unity, the priming is, in per cent., $100(1-Q)$.

44. **Records** of calorimetric tests should be even more carefully and more frequently made than in any other part of the work of a boiler-trial. The following, from work conducted by the Author, illustrates the method. The symbols relate to the first of the above formulas.

PRIMING TESTS.

| No. | Time. | Steam-pressures. | CALORIMETER. | | | | | HEAT-UNITS PER POUND FROM BOILER. | | Heat trans- ferred to Calorimeter. $W \times R = U$ | Heat from Steam. $H = T - t'$ | Heat from Water. $h = t - t'$ | Steam run into Calorimeter. x | Percentage of Priming. y |
|-----|-------|------------------|-------------------------------|----------|-------------------|-------------------|----------------|-----------------------------------------|---------------|--------------------------------------------------------------|-------------------------------------|-------------------------------------|------------------------------------------|----------------------------------|
| | | | Condens- ing Water. W | Weights. | Tempera- ture. | | | Water. t | Steam. T | | | | | |
| | | | | | Wet Steam. w | Initial. t'' | Final. t' | | | | | | | |
| | | | | | | | | | | | | | | |
| 1 | 9.46 | 21.5 | 217.6 | 14.7 | 58.9 | 123.2 | 64.3 | 262.851 | 1193.59 | 13991.68 | 1070.39 | 139.65 | 12.827 | 6.32 |
| 2 | 10.16 | 38. | 218.1 | 15.25 | 47.8 | 121.4 | 73.6 | 286.36 | 1200.61 | 16052.16 | 1079.21 | 164.96 | 14.806 | 7.29 |
| 3 | 11.15 | 40. | 250. | 16.65 | 46.7 | 122.4 | 75.7 | 288.79 | 1201.23 | 18925. | 1078.83 | 166.39 | 17.705 | 7.58 |
| 4 | 12.01 | 72.5 | 250. | 18.3 | 45.7 | 124.2 | 78.5 | 320.88 | 1210.87 | 19625. | 1086.67 | 196.68 | 18.006 | 7.77 |
| 5 | 1.01 | 61.5 | 250. | 18.2 | 44.2 | 121.6 | 77.4 | 311.27 | 1208.02 | 19350. | 1086.42 | 189.67 | 17.728 | 7.65 |
| 6 | 1.45 | 55. | 250. | 18.45 | 44.6 | 123.7 | 79.1 | 305.07 | 1209.19 | 19775. | 1082.49 | 181.37 | 18.231 | 7.82 |
| 7 | 2.25 | 60. | 250. | 18.2 | 44.2 | 122.3 | 78.1 | 309.88 | 1207.61 | 19525. | 1085.31 | 187.58 | 17.946 | 7.74 |
| 8 | 3.10 | 61.5 | 250. | 16.92 | 44.6 | 121.7 | 77.1 | 311.27 | 1208.02 | 19275. | 1086.32 | 189.57 | 17.917 | 7.68 |
| 9 | 3.55 | 65. | 250. | 17.1 | 47.3 | 121.4 | 74.1 | 314.44 | 1208.06 | 18525. | 1087.56 | 193.04 | 17.019 | 7.30 |
| 10 | 4.30 | 61. | 250. | 17.1 | 46.6 | 120.6 | 74.0 | 310.81 | 1207.88 | 18500. | 1087.28 | 190.21 | 16.996 | 7.29 |

The boiler was a water-tubular boiler, which was not so handled as to give as dry steam as was desired; and one object of the trial, of which the above is a part of the record, was to ascertain how seriously was the quality of the steam affected. It is seen that the priming amounted to seven or eight per cent, with fairly uniform figures through the period of test. The steam should have entrained less than one half this proportion, had the boiler been all that was expected of it.

Errors of small magnitude, absolutely, may greatly affect the results of calculation, as is well illustrated by the following example presented by Mr. Kent:

* Centennial Report, pp. 138-9.

Assume the values of the quantities to be, as read, column 1 :

| | OBSERVED READING. | TRUE READING. | AMOUNT OF ERROR. |
|-------------------------------------------------------------------------------|----------------------|------------------|-----------------------|
| Weight of condensing-water, corrected for equivalent of apparatus,* W | 200.5 lbs. | 200 lbs. | $\frac{1}{2}$ pound. |
| Weight of condensed steam, w | 9.9 " | 10.0 " | $\frac{1}{10}$ " |
| Pressure of steam by gauge, P | 78. " | 80 " | 2 pounds. |
| Original temperature of condensing water, t | 44°.5 " | 45° " | $\frac{1}{2}$ degree. |
| Final temperature of condensing water, t' | 100°.5 " | 100° " | $\frac{1}{2}$ " |

Then let it be assumed that errors of instruments or of observation have led to the recording of slightly different figures from the true quantities, as given in column 2 :

| Substituting in the formula the "true readings," we have for the value of | Moisture per cent. | Error per cent. |
|---------------------------------------------------------------------------|--------------------------------------|-----------------|
| All readings true except $W = 200.5$, | $Q = 0.9874 = 1.26$ | $= 0.$ |
| " " " " $w = 9.9$, | $Q = .9906 = 0.94$ | $= 0.32$ |
| " " " " $P = 78.0$, | $Q = 1.0000 = 0.00$ | $= 1.26$ |
| " " " " $t = 78.0$, | $Q = .9880 = 1.20$ | $= 0.06$ |
| " " " " $t' = 44.5$, | $Q = .9989 = 0.11$ | $= 1.15$ |
| " " " " $t' = 100.5$, | $Q = .9994 = 0.06$ | $= 1.20$ |
| " " incorrect..... | $Q = 1.0272 = (\text{minus}) = 3.98$ | |

The last case is equivalent to 50.2 degrees superheating.

Errors of 0.1 or even 0.25 per cent in weights and of temperature of equal amount not infrequently occur, probably, where ordinary instruments are employed. The errors due to false weight in measurement of the condensed steam are liable to be very serious, and it is only by making a considerable number of observations and obtaining the mean that results can be secured, ordinarily, of real value.

45. The "Coil Calorimeter" has been devised to secure more exact results in the weighing of the water of condensation than can be obtained when it is weighed as part of the larger mass. In this instrument a coil of pipe is introduced into the tank and serves as a surface-condenser in which the boiler-steam is received and condensed, and from which it is transferred to another vessel in which it is weighed by itself with scales constructed to weigh such small weights with accuracy; or the coil is removed and weighed with the contained water. In the

* Correction made only for coil calorimeter to be described.

former case, drops of water may adhere to the internal surfaces of the coil and escape measurement; in the latter, the weight to be determined is increased by the known weight of the coil, and less delicacy of weighing becomes possible.

The following is Kent's description of his calorimeter, which is of this class, and has been found to give good results: *

A surface-condenser is made of light-weight copper tubing $\frac{3}{4}$ " in diameter and about 50' in length, coiled into two coils, one inside of the other, the outer coil 14" and the inner 10" in diameter, both coils being 15" high. The lower ends of the coils are connected by means of a brazed T-coupling to a shorter coil, about 5' long, of 2" copper tubing, which is placed at the bottom of the smaller coil and acts as a receiver to contain the condensed water. The larger coil is brazed to a $\frac{3}{4}$ " pipe, which passes upward alongside of the outer coil to just above the level of the top of the coil and ends in a globe-valve, and a short elbow-pipe which points outward from the coil. The upper ends of the two $\frac{3}{4}$ " coils are brazed together into a T, and connected thereby to a $\frac{3}{4}$ " vertical pipe provided with a globe-valve, immediately above which is placed a three-way cock, and above that a brass union ground steam-tight. The upper portion of the union is connected to the steam-hose, which latter is thoroughly felted down to the union. The three-way cock has a piece of pipe a few inches long attached to its middle outlet and pointing outward from the coil.

A water-barrel, large enough to receive the coil and with some space to spare, is lined with a cylindrical vessel of galvanized iron. The space between the iron and the wood of the barrel is filled with hair-felt. The iron lining is made to return over the edge of the barrel, and is nailed down to the outer edge so as to keep the felt always dry. The barrel is furnished also with a small propeller, the shaft of which runs inside of the inner coil when the latter is placed in the barrel. The barrel is hung on trunnions by a bail by which it may be raised for weighing on a steelyard supported on a tripod and lifting lever. The steelyard for weighing the barrel is graduated

* Trans. Am. Soc. M. E. 1884.

to tenths of a pound, and a smaller steelyard is used for weighing the coil, which is graduated to hundredths of a pound.

In operation, the coil, thoroughly dry inside and out, is carefully weighed on the small steelyard. It is then placed in the barrel, which is filled with cold water up to the level of the top of the globe-valves of the coil and just below the level of the three-way cock, the propeller being inserted and its handle connected. The barrel and its contents are carefully weighed on the large steelyard; the steam-hose is connected by means of its union to the coil, and the three-way cock turned so as to let the steam flow through it into the outer air, by which means the hose is thoroughly heated; but no steam is allowed to go into the coil. The water in the barrel is now rapidly stirred in reverse directions by the propeller and its temperature taken. The three-way cock is then quickly turned, so as to stop the steam escaping into the air and to turn it into the coil; the thermometer is held in the barrel, and the water stirred until the thermometer indicates from five to ten degrees less than the maximum temperature desired. The globe-valve leading to the coil is then rapidly and tightly closed, the three-way cock turned to let the steam in the hose escape into the air, and the steam entering the hose shut off. During this time the water is being stirred, and the observer carefully notes the thermometer until the maximum temperature is reached, which is recorded as the final temperature of the condensing water. The union is then disconnected and the barrel and coil weighed together on the large steelyard; the coil is then withdrawn from the barrel and hung up to dry thoroughly on the outside. When dry it is weighed on the small scales. If the temperature of the water in the barrel is raised to 110° or 120° the coil will dry to constant weight in a few minutes. After the weight is taken, both globe-valves to the coil are opened, the steam-hose connected, and all of the condensed water blown out of the coil, and steam allowed to blow through the coil freely for a few seconds at full pressure. When the coil cools it may be weighed again, and is then ready for another test.

If both steelyards were perfectly accurate, and there were no losses by leakage or evaporation, the difference between the

original and final weights of the barrel and contents should be exactly the same as the difference between the original and final weights of the coil. In practice this is rarely found to be the case, since there is a slight possible error in each weighing, which is larger in the weighing on the large steelyard. In making calculations the weights of the coil on the small steelyard should be used, the weight on the large steelyard being used merely as a check against large errors.

The late Mr. J. C. Hoadley constructed exceedingly accurate apparatus of the "coil" type and obtained excellent results.

It is evident that this calorimeter may be used continuously, if desired, instead of intermittently. In this case a continuous flow of condensing water into and out of the barrel must be established, and the temperature of inflow and outflow and of the condensed steam read at short intervals of time.

46. The Continuous Calorimeter is an instrument in which the operations of transfer of steam to the instrument and its examination are not intermitted, as is necessarily the case in the more commonly employed forms of the apparatus. The instrument* being thus kept in use continuously, every variation in the quality of steam can be observed and the number of observations can be increased to any desired extent, and, the apparatus being accurate, any degree of exactness of mean results can be attained.

One of the earliest forms of this instrument was devised by Mr. John D. Van Buren, of the U. S. N. Engineers, and Instructor in Engineering at the Naval Academy, about 1867. This instrument, as constructed by Mr. T. Skeel, and used by a committee of judges* at the exhibition of the American Institute, 1874-5, of which the Author was chairman, was made as follows:

Steam was drawn from the steam-drum, near the safety-valve, through a felted pipe $1\frac{1}{2}$ inches (3.8 cm.) diameter, into a rectangular spiral or coil consisting of 80 feet (24.4 m.) of pipe of similar size. Condensing water from the street-main was led into the tank surrounding the coil or "worm," and

* Trans. Am. Inst. 1875; Van Nostrand's Mag. 1875.

issued at the bottom through a "standard orifice," the rate of discharge from which had been determined and the law of its variation with change of head ascertained. The quantity of condensing water thus became known by observing the head of water within the tank. The water of condensation from the coil was caught in a convenient vessel, and weighed on scales provided for that purpose. The temperature of the condensing water at entrance and exit was shown by fixed thermometers, and that of the water of condensation at its issue from the coil was similarly shown, while the steam-gauge placed on the boiler gave the other needed data. The calculations are evidently precisely the same as with the preceding type of calorimeter.

The Barrus Calorimeter * (Fig. 2) is essentially of a small surface-condenser. The steam enters by the pipe *j*. The con-

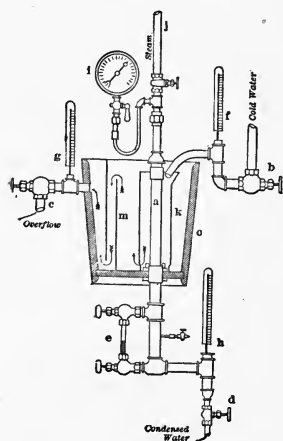


FIG. 2.—THE CONTINUOUS CALORIMETER.

densing-surface, *a*, is a continuation and enlargement of the supply-pipe, a 1-inch (2.54 cm.) iron pipe with a length of 12 inches (30.4 cm.) of exposed surface. This pipe is under the full pressure of steam. The condensed water collects in the lower parts of the apparatus, where its level is shown in the glass, *e*, and is drawn off by means of the valve, *d*. The injection-water, cooled to a temperature of 40° Fahr., or less, enters the wooden vessel, *a*, through the valve, *b*, and circulates around the condensing pipe, carried downward to the bottom by means of the tube *k*, and overflows at the pipe, *c*, after passing through the mixing chambers, *m*. The amount of water admitted is regulated so as to secure a temperature at the overflow of 75° or 80° Fahr., or the approximate temperature of the surrounding atmosphere. The thermometers, *f* and *g*, which are read to

* Trans. Am. Soc. M. E. 1884.

tenths of a degree, show the temperature of injection and overflow water, and the thermometer, *h*, shows that of the condensed water. The overflow water and the condensed water are collected in a system of weighing tanks. The steam-pipe down to the surface of the water, and the pipes in the lower part of the apparatus, are covered with felt.

There is no wire-drawing of the steam, and no allowance to be made for specific heat of the apparatus. The only correction to be made of material amount is for radiation from the pipes covered with felt, and this can be accurately determined by an independent radiation experiment, made when the condenser vessel is empty.

Another form of instrument devised by the same engineer is arranged in such manner as to permit the steam from the boiler to be dried and the quantity of heat so employed measured as a gauge of the amount of water contained in the steam. This form of this apparatus is found very satisfactory.*

The pipe conveying the steam to be tested is usually a half-inch (1.27 cm.) iron pipe. A long thread is cut on this pipe, and it is screwed into the main steam supply-pipe of the boiler in such a manner as to extend diametrically across to the opposite side. The inclosed part is perforated with from 40 to 50 small holes, and the open end of the pipe sealed. If the pipe is screwed into the under side the perforations begin at a distance of one inch (2.54 cm.) from the bottom. The connection is made as short as possible, and covered with felt. Where the calorimeter can be attached to the under side of the main, the distance to the top valve need not exceed six inches (15 cm.). In this position it is self-supporting. The steam for the superheater is also supplied by a half-inch iron pipe, but this may be attached to the main at any convenient point.

Steam to be tested enters by the pipe, which has a jacket. On passing out the thermometer gives its temperature, and it is discharged through a small orifice $\frac{1}{8}$ inch (0.32 cm.) in diameter. Steam to be superheated enters and is superheated by a gas-lamp, passes the thermometer,

*Trans. Am. Soc. Mech. Engrs., vol. vii. p. 178.

and issues through an opening like that for the steam. The thermometers are immersed in oil-wells surrounded by the current of steam to be tested, or of that used in drying the boiler-steam.

In the operation of this calorimeter steam at full pressure enters the apparatus, and the jacket-steam is heated until a perceptible rise of temperature above that due the pressure indicates that its moisture has been evaporated. The working having become steady, the difference between the temperatures is noted and corrected by deducting the excess above that of moist steam at the observed pressure, and the number of degrees of superheating thus determined, as the rate of flow is the same from both orifices. Here the evaporation of one per cent of moisture from steam at 80 pounds pressure (5.6 kilogs. per sq. cm.) reduces the temperature of superheated steam about $18^{\circ}.7$ Fahr. ($10^{\circ}.4$ Cent.), and the percentage of moisture is obtained by dividing the range of superheat, as above, by this number, or generally by the quotient of the latent heat at the observed pressure by 47.5. The following are data and results obtained by the use of this apparatus:

DATA AND RESULTS IN FULL OF CALORIMETER TESTS.

| Number for Reference. | Date. | Gauge-pressure. | Number of degrees inlet steam was superheated. | Number of degrees outlet steam was superheated. | Number of degrees wet steam was superheated. | Number of degrees lost by superheated steam due to radiation from calorimeter. | Number of degrees representing radiation from supply-pipe. | Amount of Moisture in the Wet Steam. | |
|-----------------------|---------|-----------------|------------------------------------------------|-------------------------------------------------|----------------------------------------------|--------------------------------------------------------------------------------|------------------------------------------------------------|--------------------------------------|---------------------------|
| | | | | | | | | Expressed in degrees of superheat. | Expressed in percentage.* |
| 1 | Apr. 13 | 89. | 99. | 54.5 | 8. | 8. | 9.5 | 19. | 1.02 |
| 2 | " 14 | 89. | 75. | 37. | 5.5 | 8. | 9.5 | 16. | 0.86 |
| 3 | " 15 | 86. | 74. | 37. | 7. | 10.5 | 9.5 | 10. | 0.54 |
| 4 | " 16 | 86. | 74. | 39. | 9.5 | 7. | 9.5 | 9. | 0.49 |
| 5 | " 30 | 85. | 72. | 38. | 10.5 | 8. | 9.5 | 6. | 0.32 |
| 6 | May 4 | 80. | 77.5 | 41.5 | 9.5 | 8. | 9.5 | 9. | 0.49 |
| 7 | " 5 | 84. | 68. | 36.5 | 6.5 | 7.5 | 9.5 | 8. | 0.43 |

NOTE.—The duration of each of these tests was about one hour.

* Obtained by dividing the preceding column by 18.6, the number of degrees corresponding to 1 per cent of moisture.

An exceedingly simple form of calorimeter, practically available when the steam is fairly dry, is that devised by Professor Peabody, which depends on the fact that dry steam is superheated by wire-drawing.*

A piece of pipe six inches in diameter and ten inches long was capped at each end. Into the upper end was fitted a half-inch pipe bringing the steam to be tested, a thermometer cup, and a steam-gauge. From the lower cap an inch pipe led away the exhaust steam. Near the calorimeter was a T which formed a pocket, with a drip at the lower opening, and a branch from the side opening leading to an angle valve in the upper cap of the condenser. The pipe further was well wrapped with hair felt, and the calorimeter was wrapped in asbestos board and hair felt, and covered with russia iron.

Two other calorimeters differ from the first only in size. One is made of a piece of two-inch pipe eight inches long, and the other of a piece of four-inch pipe of the same length. The smaller are more sensitive.

To make an experiment, the valve in the supply-pipe is partly opened, and a valve in the exhaust-pipe is regulated to give the desired pressure in the calorimeter. After the gauge and thermometer attached become steady, their readings are taken, and the reading of the boiler-gauge.

If p_1 is the boiler-pressure, l is the heat of vaporization, and h the heat of the liquid corresponding, x may represent the dry steam in one pound of the mixture from the steam-pipe; $1 - x$ is the water or priming. The heat in one pound of the mixture is

$$xl + h.$$

Let p_2 be the pressure in the calorimeter, and h_1 the total heat, and t_1 the temperature corresponding. Let t_2 be the temperature of the superheated steam by the thermometer. Then the heat in one pound of steam in the calorimeter is

$$h_1 + c_p(t_2 - t_1),$$

* Trans. Am. Soc. M. E., 1888, vol. x.

in which c_p is the specific heat of the superheated steam at constant pressure (0.4808).

Assuming that no heat is lost,

$$xl + h = h_1 + c_p(t_2 - t_1);$$

$$\therefore x = \frac{h_1 + c_p(t_2 - t_1)}{l} - h,$$

and the priming is

$$1 - x.$$

The following experiments were made:

| GAUGE PRESSURES. | | Temperature in the calorimeter F. | Priming. |
|------------------|--------------|-----------------------------------|----------|
| Boiler. | Calorimeter. | | |
| 71.2 | 38.5 | 286.7 | 0.011 |
| 60.3 | 26.8 | 271.8 | 0.012 |
| 63.0 | 17.5 | 264.9 | 0.013 |
| 60.6 | 7.0 | 258.8 | 0.011 |
| 69.0 | 3.7 | 258.1 | 0.012 |

The other calorimeters gave substantially the same results.

This type of calorimeter can be used only when the priming is not excessive; otherwise the wire-drawing will fail to superheat the steam. To find this limit for any pressure, we may assume that the steam is just dry and saturated at that limit in the calorimeter. The limit is higher for higher pressures, but the calorimeter can be applied only where the priming is moderate, thus:

| PRESSURE. | | Priming. |
|-----------|--------|----------|
| Absolute. | Gauge. | |
| 300 | 285.3 | 0.077 |
| 250 | 235.3 | 0.070 |
| 200 | 185.3 | 0.061 |
| 175 | 160.3 | 0.058 |
| 150 | 135.3 | 0.052 |
| 125 | 110.3 | 0.046 |
| 100 | 85.3 | 0.040 |
| 75 | 60.3 | 0.032 |
| 50 | 35.3 | 0.023 |

The limit may be extended by connecting the exhaust to a condenser. The limit at 100 pounds absolute, with 3 pounds absolute in the calorimeter, thus becomes 0.064, instead of 0.046.

The thermometer should be absolutely reliable. A considerable error in the temperature would produce an inconsiderable effect on the result in other cases. Thus, at 100 pounds absolute with atmospheric pressure in the calorimeter, 10° F. of superheating indicates 0.035 priming, and 15° F. indicates 0.032 priming. A slight error in the gauge-reading has little effect. If reading be apparently 100.5 pounds absolute instead of 100, with 10° of superheating, the priming appears to be 0.033 instead of 0.032.

In the Barrus calorimeter, as has been seen, the steam to be tested is dried and superheated by a stream of highly superheated steam. The following table has been calculated on the assumption that the superheated steam has an initial temperature of 500°, and a final temperature of 10° above the temperature of saturated steam of the given pressure, while the moist steam is supposed to be dried and superheated 5°. The limit under these conditions is widest for lowest pressures, and also is narrower at high pressures than that of the new type :

| PRESSURE. | | Priming. |
|-----------|--------|----------|
| Absolute. | Gauge. | |
| 50 | 35.3 | 0.170 |
| 75 | 60.3 | 0.095 |
| 100 | 185.3 | 0.086 |
| 125 | 110.3 | 0.078 |
| 150 | 135.3 | 0.071 |
| 175 | 160.3 | 0.065 |
| 200 | 185.3 | 0.059 |
| 250 | 235.3 | 0.049 |
| 300 | 285.3 | 0.040 |

One or another of these instruments may thus be best, according to pressure of steam carried.

Many other forms of calorimeter have been devised, but space will not permit their description.

47. The Analysis of Gases* issuing from the furnace and passing up the chimney is sometimes an important detail of the work of testing a steam-boiler. Such an investigation involves only an operation of great simplicity which can easily be performed by any engineer. If it is not found convenient to make the analysis in the office of the engineer, he can have the work done, at little expense, by a chemist of known skill and reliability. It is only by a knowledge of the proportions of constituents of the flue-gases that it can be determined whether the combustion is complete, whether the products of combustion are diluted with excess of air, and whether the fuel used has been so burned as to give its best effect. Such analyses also enable the engineer to ascertain the best method of burning the fuel.

In sampling the gases, a matter in regard to which some precaution is advisable, the method of Mr. Hoadly is found very satisfactory.†

Very great diversities in composition often exist in the same flue at the same time. To obtain a sample, allow one orifice to draw off gases through for each 25 sq. inches (161 sq. cm.) of cross-section of flue. The pipes must be of equal diameter and of equal length. These should be secured in a box of galvanized sheet-iron, equal in thickness to one course of brick, so that the ends may be evenly distributed over the flue *A* (Fig. 3), and their other open ends inclosed in the

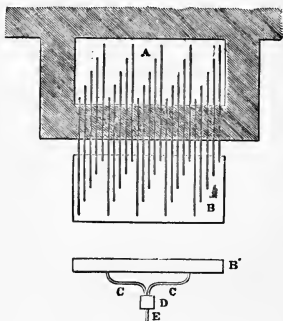


FIG. 3.—FLUE-GAS SAMPLING.

* Consult Handbook of Gas Analysis, by C. Winkler. London: J. Van Voorst. 1885.

† Trans. Am. Soc. M. E., vol. vi.

receiver *B*. If the flue gases be drawn off from the receiver *B* by four tubes *CC*, into a mixing box *D*, beneath, a good mixture can be obtained.

The sampling of the gas should be carried out at intervals of 10 to 15 minutes throughout the trial. The gas should be received in an air-tight pipe or jar. The composition of the gases should be determined as far as regards carbonic acid, carbonic oxide, and oxygen. The tube should be of porcelain or glass for very hot flues, since iron tubes at such temperatures are oxidized. Supposing an analysis of the gas give *K* per cent of carbonic acid, *O* per cent of oxygen, and *N* per cent of nitrogen, then the proportion of air actually used to the theoretical quantity required is 1 to *x*.

Where

$$x = \frac{N}{N - \frac{79}{21}O} \text{ or } \frac{21}{21 - 79\frac{O}{N}}$$

unity of weight of this coal will then give, at a temperature of 0° and a pressure of one atmosphere,

$$\frac{1854}{10}C = \text{carbonic acid};$$

$$\frac{KO}{K} = \text{oxygen};$$

$$\frac{KN}{K} = \text{nitrogen}.$$

The quantity of moisture in the escaping gases may be calculated from the moisture in the coal, from that formed by burning the hydrogen, and from that contained in the air admitted to the furnace where the latter has been determined. Any serious break in the setting can be detected by filling the grate with smoky coal and then closing the damper.

The following sketch shows the apparatus employed by Mr. Wilson.*

* Journal Society of Arts, Feb. 1889.

A. Apparatus employed for the gas analyses. The whole apparatus being filled with mercury, the gas is introduced into the eudiometer *a* and its volume measured. The stopcock *b* and the three-way cock *c* are then opened, and the gas passes over into the laboratory vessel *d*, followed by some mercury to drive all the gas out of the capillary tube. The reagent is then poured into the cup *e*, and admitted to the laboratory vessel by the three-way cock. When the absorption is complete, the mercury bottle is placed on the upper shelf and the cocks being opened the gas passes back into the eudiometer. When the reagent rises to *c* the three-way cock is turned to communicate with the cup so that the reagent passes into it. Some mercury is then driven over into the eudiometer to clear the gas from the capillary tube, and the volume is again read.

The two ends of the capillary tubes at *f* are made funnel-shaped, and connected by a thick india-rubber tube. By lowering the eudiometer a little when the gas is passed from *a* to *d*, and raising it for the passage in the opposite direction, the whole of the gas is driven out of the tube.

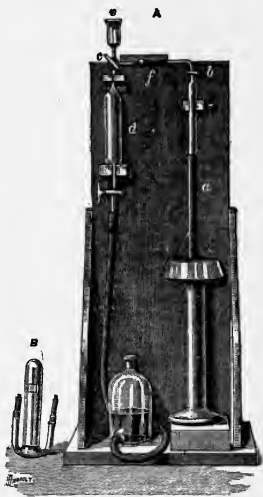


FIG. 4.—APPARATUS FOR GAS ANALYSIS.

B. One of the tubes used for taking samples of gas. The sampler, completely filled with mercury, is connected with the gas to be taken by means of an india-rubber tube (previously aspirated if necessary). The vessel is then inclined so as to allow mercury to flow out of the opposite tube until only enough remains to seal up the sample.

The apparatus designed by Professor Elliott, and employed in work carried on under the direction of the Author, consists,

as shown in Fig. 5, of two vertical glass tubes, AB , $A'B'$, joined by rubber-tubing, E , at their upper ends. The large tube, AB , is the treating, the smaller, $A'B'$, the measuring tube; the latter is suitably graduated to cubic centimetres. Water-bottles, K , L , are connected with the lower ends of the tubes by tubing, NO , $N'O'$, and are used in effecting transfer of the gas from tube to tube. M is a funnel through which the reagents used may be introduced. G , F , and I are cocks of suitable size and construction.

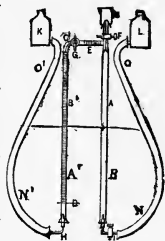


FIG. 5.—APPARATUS FOR GAS ANALYSIS.

In filling the apparatus it is set up conveniently near the flue, and the line of tubing from the collector, within the latter, is connected with the tube AB . The receiver L being detached the lower end of AB is connected with an aspirator or equivalent apparatus, such, for example, as might be improvised by the use of an air-tight tank or a barrel; and the flow thus produced, when the aspirator is emptied of its water, fills the tube AB with gas drawn from the flue. It is retained by closing the valves F and I , which had been open during the operation of filling. The tube is then disconnected from the aspirator, and the receiver, or bottle, L , connected as shown, and in such manner that no air can reach the tube AB .

Removing the apparatus to the laboratory or other convenient location, the analysis is made as follows:

Pass into $A'B'$ a convenient volume, as 100 c.c. of the gas, and discharge the remainder through the valve and funnel F and M , filling the tube AB with water from L . Transfer the measured gas back to AB , through E , and add a solution from M , which will absorb some one constituent. Return the gas to $A'B'$, and again read its volumes. The difference is the quantity of gas absorbed. Repeat this process, using next an absorbent which will take up a second constituent of the gas, and thus obtain a second measure of volume; and thus continue until all the desired determinations are made. All readings should be made at the same temperature, or practically so. The tube

AB should be well washed at each operation, in order that no reagent should be affected by traces of that previously used.

The absorbents employed are best taken in the following order:

1. Caustic potash—to absorb carbonic acid.
2. Potassium pyrogallate—to absorb free oxygen.
3. Cuprous chloride in concentrated hydrochloric-acid solution—to absorb carbonic oxide.

After their use nitrogen will remain, and will be measured as a balance which, added to the sum of the measured volumes of gases absorbed, should give the original total. Where weights are to be determined, the volumetric measures obtained as above are to be reduced by the usual process.

The atomic weights of the principal constituents being, oxygen, 16; nitrogen, 14; carbon monoxide, 28; carbon dioxide, 44, we shall have by percentages, where the symbols represent per cent in volumes, for each, when the total is

$$M = 14N + 16O + 28CO + 44CO_2,$$

$$\frac{14N}{M}, \quad \frac{16O}{M}, \quad \frac{28CO}{M}, \quad \frac{44CO_2}{M}, \text{ respectively.}$$

Since the total per cent of oxygen is measured by $\frac{32}{44}CO_2 + \frac{16}{28}CO + \text{free oxygen}$, and the total per cent of carbon is $\frac{12}{44}CO_2 + \frac{12}{28}CO$, we shall have for the percentage of each,

$$O' = \frac{32 \times 44 \times CO_2}{44M} + \frac{16 \times 28 \times CO}{28M} + \frac{16O}{M};$$

$$C' = \frac{12 \times 44 \times CO_2}{44M} + \frac{12 \times 28 \times CO}{28M};$$

or,

$$O' = 32 \frac{CO_2}{M} + 16 \frac{(CO + O)}{M};$$

$$C' = 12 \frac{CO_2}{M} + 12 \frac{CO}{M}.$$

The total oxygen is that which entered the furnace as the supporter of combustion, and is a measure of the air supplied. The ratio of free to combined oxygen is a measure of the ratio of the air acting as a diluent simply to that supporting combustion.

Thus these measurements exhibit the efficiency of combustion, the quantity of air employed, and the magnitude of the wastes of heat at the chimney, occurring through imperfect combustion or excess of air-supply. It is evident, however, that where moisture or steam accompanies the gases, it escapes measurement; this, however, introduces no important error in ordinary work.

Efficiency of combustion is indicated by the analysis of the flue-gases with very great certainty. The appearance of carbon monoxide at the chimney proves the combustion to be imperfect in proportion as it is more or less abundant. The presence of unconsumed oxygen, on the other hand, in the absence of carbon monoxide, proves an excess of air-supply. Both gases appearing is a proof of incomplete intermixture of air and combustible, or of so low a temperature of furnace as to check combustion. This analysis being compared with that of the fuel reveals the character and the perfection of combustion, and permits a very exact determination to be made of the specific heat of the gases, and is thus a check on calculations of wasted heat.

48. Draught-gauges are made for the purpose of determining the head-producing draught and the intensity of the draught, which are of many forms, but which usually depend upon the measurement of the head of water which balances that head at the chimney. A very compact and accurate form

of draught-gauge, used by the Author with very satisfactory results, is that of Mr. J. M. Allen (Fig. 6).

A and *A'* are glass tubes, mounted as shown, communicating with each other by a passage through the base, which may be closed by means of the stop-cock shown. Surrounding the glass tubes are two brass rings, *B* and *B'*. These rings are attached to blocks which slide in dovetailed grooves in the

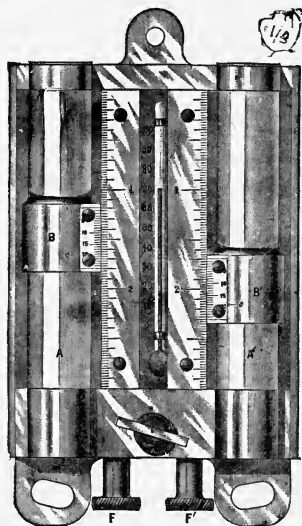


FIG. 6.—DRAUGHT-GAUGE.

body of the instrument, and may be moved up and down by screws at *FF'*. The scales are divided into fortieths of an inch, and read to thousandths of an inch by the verniers *e* and *e'*, which are attached to the sliding rings *B B'*. If the two short rings are set at different heights, the difference in readings will give the difference of level. The thermometer is for the purpose of noting the temperature of the external air. The method of using the instrument is as follows:*

* *The Locomotive*, May, 1884, p. 67.

venient point near the base of the chimney a hole is made large enough to insert a thermometer. The height from this opening to the top of chimney, and also of grates, should be noted. The chimney-gauge is attached to some convenient wall. The tubes are filled about half full of water, when the verniers afford an easy means of setting it perpendicular. One end of a flexible rubber tube is then inserted into the upper end of one of the glass tubes, and the other end of the tube is in the chimney-flue. The tubes *BB'* are adjusted until their upper ends are just tangent to the surface of the water in the two tubes. The reading of the two scales is then taken, and their difference. At the same time the temperature of the flue is noted, as well as that of the external atmosphere. Comparison may then be made with the following table, computed for use in this connection for a chimney 100 feet high, with various temperatures outside and inside of the flue, and on the supposition that the *temperature of the chimney is uniform from top to bottom*—an inaccurate though usual assumption, however. For other heights than 100 feet, the theoretical height is found by simple proportion, thus: Suppose the external temperature is 50° , temperature of flue 380° , height of chimney 137 feet, then under 60° at the top of the table, and opposite to 380° interpolated in the left-hand margin, we find .52".

Then $100 : 137 :: .52'' : .71''$, which is the required height for a 137-foot chimney, and similarly for any other height.

HEIGHT OF WATER COLUMN DUE TO UNBALANCED PRESSURE IN CHIMNEY 100 FEET HIGH.

| Temperature in the Chimney. Fahr. | TEMPERATURE (FAHR.) OF THE EXTERNAL AIR—BAROMETER, 14.7. | | | | |
|--------------------------------------------|----------------------------------------------------------|------|------|------|------|
| | 20° | 40° | 60° | 80° | 100° |
| 220 | .419 | .355 | .298 | .244 | .192 |
| 250 | .468 | .405 | .347 | .294 | .242 |
| 300 | .541 | .478 | .420 | .367 | .315 |
| 350 | .607 | .543 | .486 | .432 | .380 |
| 400 | .662 | .598 | .541 | .488 | .436 |
| 450 | .714 | .651 | .593 | .540 | .488 |
| 500 | .760 | .697 | .639 | .586 | .534 |

The most common form of gauge-testing apparatus is shown in the accompanying engraving. The standard gauge, which is known by comparison with a mercury column or by other

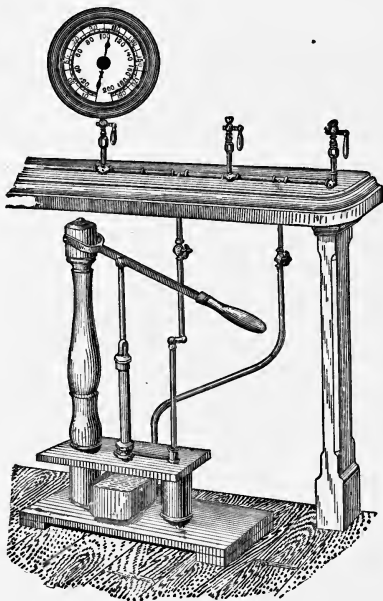


FIG 7.—GAUGE-TESTING PUMP.

test to be right, is mounted as shown. The instrument to be tested is attached to one of the other cocks, and, both being subjected to the same pressures, a comparison of their readings will exhibit the errors of the second gauge.

49. A Sample Trial is described in the following report, which will illustrate well the methods and results of a carefully made test of a boiler in which a complete trial was attempted, under the direction of the Author.*

* Sci. Am. Supplement, No. 641, p. 10234.

Trial of a Water-tube Boiler.

This boiler was used to supply steam to one or more engines, as needed, or to heat the buildings of the college. The principal dimensions are as follows:

| | |
|---------------------------------------------------------------|----------------|
| Length of drum, | 13 ft. |
| Diameter, | 2 ft. 6 in. |
| Number of water-tubes, | 40 |
| Outside diameter of tubes, | 4 in. |
| Length, | 13 ft. 8 in. |
| Width of furnace, | 3 ft. 3½ in. |
| Length of furnace, | 6 ft. 1 in. |
| Length of grate-bars, | 3 ft. |
| Width of grate-bars, | ¾ in. |
| Width of air-spaces, | ¾ in. |
| Number of grate-bars, | 54 |
| Area of chimney, | 3.65 sq. ft. |
| Height of chimney, | 60.25 ft. |
| Area of grate-surface, | 20 sq. ft. |
| Area of heating-surface, | 682.57 sq. ft. |
| Area for draught between tubes, | 4.75 " |
| Ratio of grate to heating-surface, | 1 : 34.1 |
| Ratio of draught area to grate, | 0.25 |
| Ratio of grate-surface to cross-section of chimney, | 5.48 |
| Ratio of area of grate to area of air-spaces, | 2.24 |
| Whole area of damper opening, | 3 sq. ft. |

The main steam-pipe after passing horizontally to the rear of the "setting" descends vertically a distance of 4 ft. and passes out of the boiler-room to the chimney. Draught is produced by a chimney which rises directly at the back end of the boiler, the first 9½ ft. being brick and the remainder a sheet-iron cylindrical stack. A vertical sliding damper is placed in the opening leading to the chimney. Two partitions of fire-

brick supported by iron plates are placed transversely across the nest of water-tubes. The first is 7 ft. 1 in. from the front end of the tubes, and the second 3 ft. 7 in. from the first. These partitions cause the gases to pass among the tubes three times, then across the rising tubes into the back connection, and from there to the chimney.

The object of the trial was: 1. To determine the evaporative efficiency of the boiler. 2. To estimate the horse-power developed under ordinary working conditions, a horse-power being taken as equivalent to 30 lbs. of feed-water supplied per hour at a temperature of 100° F., and evaporated under 70 lbs. gauge-pressure.

Previous to the test all cracks and holes in the setting and around the doors leading to the flues were carefully stopped with fire-clay and mortar. The blow-off and return "drip" pipes were disconnected and caps placed on the exposed ends. An injector feed-pipe connected with the boiler was left in place, as its disconnection would be attended with some difficulty. The overflow-pipe was, however, left open, in order to detect any leak which might occur. The feed-pipe was disconnected from the "mains," and a suction-pipe from it placed in a barrel into which the feed-water was run after having been weighed. A pipe leading to the outside of the boiler-house was connected with the main steam-pipe, so that all steam made by the boiler, over and above that required to run the engine and heat the buildings, could be discharged into the air.

At 7 A.M. April 28, the fire, which had been banked on the preceding evening, was started, and the steam pressure brought to 80 lbs. by the large gauge. The fire was then quickly drawn and the contents of the ash-pit removed. A new fire was started immediately with a weighed quantity of hemlock wood and brought to the normal condition with coal. The amount of water shown by the water-glass was noted. At 8 A.M. the engine was started, and the trial commenced. Both ash-pit doors

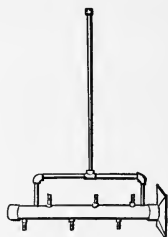


FIG. 8.—GAS SAMPLER.

were left open at first and the damper wide open. The damper was lowered 3 in. at 9.30 A.M., and at 12.50 a further amount of 3 in. At 11.17 A.M. one of the ash-pit doors was closed and so remained during the remainder of the trial. The effect of this arrangement of damper and draught door was observed in the higher temperature of the flue gases at the base of the chimney.

The fuel used was anthracite coal, known in the market as "grate coal." An average sample of this coal was weighed, pulverized, and placed in an evaporating oven to dry. After seven hours it was found to have lost 3.81 per cent. in weight. In



FIG. 9.—PYROMETER.

working up the results of the trial, this figure was taken to represent the percentage of moisture in the coal. The coal was weighed by the barrow load in uniform charges of 200 lbs. each, and dumped before the door as needed.

The stoking or firing was performed regularly every half hour and fire cleaned every third time. During the period of stoking, the back damper was closed to avoid loss of heat by the current of cold air which otherwise would rush through the heated flues. The feed-water was drawn from the mains into a barrel placed on a platform scale, where it was carefully weighed. It was then drawn off into another barrel, from which it was pumped into the boiler by a steam-pump of the ordinary type. It was the endeavor to deliver the water to the boiler as continuously as possible. The temperature of the feed-water was noted at each weighing.

The observations which were called for, and the results of which were finally recorded as the log of the trial, were such as would ordinarily be demanded in any usual case of engineering practice of this character, and were sufficient to enable the observers to make all essential computations; while none were made which were not either of importance in that respect, or of real interest to the engineer concerned in the questions proposed to be settled by the trial as conducted.

The method of conduct of the trial, in tolerably full detail, is described in the succeeding pages. The observations made are indicated; the processes involved in their reduction, computation, and tabulation are exhibited and illustrated; and the final deductions and conclusions are stated at length:

The following observations were made every half-hour:

1. Temperature of flue gases at the base of chimney.
2. Temperature of boiler-room.
3. Temperature of outside air.
4. Reading of draught pressure-gauge.
5. Readings of the several steam-gauges.

6. The pyrometer used in measuring the temperature of the flue gases had previously been compared with a mercury thermometer between temperatures of 213° F. and 322° F. This was accomplished by means of a simple apparatus shown in Fig. 9. The stem of the pyrometer was inclosed in a steam-pipe which has communication to the boiler through a smaller pipe fitted with a stop-valve. The thermometer used in the comparison was also screwed into the larger pipe. As steam was admitted the mercury rose, and soon registered a temperature corresponding to the steam-pressure, which was kept constant for several minutes until the pyrometer reading no longer changed.

Both readings were noted, and more steam admitted, giving a higher temperature.

The several readings were "plotted," and the law of variation of the pyrometer from the thermometer reading was found to be approximately a straight line, continually falling below and diverging from the line representing the temperature as read from the thermometer.

The pyrometer was corrected from this line, and is believed to be approximately correct.

The draught pressure-gauge, which was attached to the stack near the base, was made for the Sibley College laboratories by the Hartford Steam Boiler Insurance Co. It consisted of a U-tube partially filled with water and provided with a movable vernier and scale for measuring the difference in level of the water in the two arms.

A recording steam-gauge and a mercury-gauge were attached to the boiler in addition to the large gauge ordinarily used.

The mercury-gauge was taken as the standard, and the others corrected by it.

Experiments were made every hour to determine the quality of the steam. A well-made barrel which had been thoroughly shellacked inside was placed on a very sensitive standardized platform scale, made for this work, the beam of which was graduated to $\frac{1}{20}$ of a pound and provided with a sliding poise.

The steam was taken from the main steam-pipe, 1 ft. from its connection with the boiler, and was conducted to the calorimeter through a $\frac{1}{2}$ -in. pipe, 9 ft. long, to the end of which was attached a piece of rubber hose 7 ft. long. The pipe was covered with hair-felt to prevent radiation of heat. Before placing the end of the hose in the calorimeter, steam was allowed to blow through until all the water of condensation had been discharged and the pipe and hose were thoroughly warmed up. The end of the hose was given an inclination downward toward the bottom of the barrel by means of a light strip of wood fastened to it. The steam passing into the condensing water at an angle produced a strong agitation, and thus a thorough mixture of the water was effected.* A standard Centigrade thermometer, graduated to tenths of one degree, was used with the calorimeter, and the readings were afterward reduced to the Fahrenheit scale.

During the trial five samples of flue gas were taken for analysis. The tabulated results of the analysis are as follows:

PER CENT. BY VOLUME.

| No. | Time. | CO ₂ , Observed. | Free O ₂ , Observed. | CO, Calculated. | N, Calculated. |
|-----|------------|--------------------------------|------------------------------------|--------------------|-------------------|
| 1 | 8.30 A.M. | 12 | 5.2 | 4.6 | 78.13 |
| 2 | 10.20 A.M. | 12 | 6.7 | 2.16 | 79.13 |
| 3 | 12.20 P.M. | 11.1 | 7.9 | 1.6 | 79.3 |
| 4 | 2.20 P.M. | 11.7 | 6.8* | 2.5 | 79 |
| 5 | 4.20 P.M. | 11.5 | 7 | 2.5 | 79 |

* Probably, on the whole, as good an arrangement as any plan involving the use of stirring apparatus. See Manual of the Steam Boiler.

BY WEIGHT.

| No. | Time. | CO ₂ , Calculated. | Free O, Calculated. | CO, Calculated. | N, Calculated. |
|-----|------------|----------------------------------|------------------------|--------------------|-------------------|
| 1 | 8.30 A.M. | 17.56 | 5.5 | 4.33 | 72.62 |
| 2 | 10.20 A.M. | 17.52 | 7.07 | 2 | 73.40 |
| 3 | 12.20 P.M. | 16.27 | 8.39 | 1.95 | 73.86 |
| 4 | 2.20 P.M. | 17.11 | 7.19 | 2.32 | 73.38 |
| 5 | 4.20 P.M. | 16.83 | 7.41 | 2.32 | 73.44 |

| No. | Time. | Per Cent by Weight, Total O. | Per Cent by Weight, Total C. | Air Supplied, Per lb. C. | Free O Combined O |
|-----|------------|------------------------------------|------------------------------------|--------------------------------|----------------------|
| 1 | 8.30 A.M. | 20.74 | 6.65 | 14 | 0.36 |
| 2 | 10.20 A.M. | 20.96 | 5.64 | 16.7 | 0.51 |
| 3 | 12.20 P.M. | 21.31 | 5.27 | 18 | 0.64 |
| 4 | 2.20 P.M. | 20.95 | 5.66 | 16.6 | 0.52 |
| 5 | 4.20 P.M. | 20.97 | 5.58 | 16.9 | 0.54 |

Professor Elliott's apparatus, Fig. 10, was used for the analysis. For the absorption of CO₂, a solution of potassic hydrate (1 to 20) was used, and for oxygen absorption, potassic pyrogallate; this latter being prepared by adding 5 per cent of pyrogallic acid to a solution of potassic hydrate (1 to 8). Numbers 1 and 2 were tested for CO with cuprous chloride, but as none was absorbed, and it was evidently present, the amount was calculated as follows:

For No. 1 we have 12 per cent CO₂, whose volume is equal to the volume of the O which combined to form it, and 52 per cent of free O. The volume of O in these two is, therefore, = 12 + 5.2 = 17.2 per cent. Assuming that the atmospheric air is composed of 4 parts of N and 1 part of O, by volume, to correspond to this 17.2 per cent of O we should have 17.2 × 4 = 68.8 per cent N; but after absorbing the 17.2 per cent of CO₂ and O, there remains 100 - 17.2 = 82.8 per cent. Taking 68.8 per cent from 82.8 per cent, we have 14 per cent, which must be composed of N and CO. Since the volume of CO is equal to twice the volume of the combined O, we shall have the volume of O = $\frac{CO}{2}$, and since there is four times as

much N as O, the $N = \frac{4CO}{2} = 2CO$. Therefore of this 14 per cent. 1 part is CO and 2 parts are N; $\therefore CO = \frac{14}{3} = 4.6 +$, and $N = 4.6 \times 2 = 9.3 +$, which being added to the 68.8 per cent. N, which corresponds to the free O, and that of the CO,

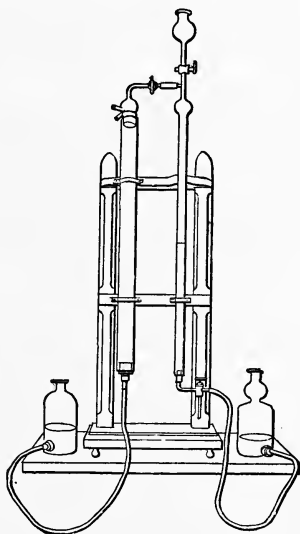


FIG. 10.—GAS ANALYSIS.

= 78.13 per cent. To reduce per cent. by volume to per cent. by weight, we use the following constants:

| | | | | | |
|------------------------------------------------------|---|---|---|-----|---------|
| Weight of 1 liter of CO ₂ , 1.9774 grams. | | | | | |
| " | " | 1 | " | O, | 1.43 " |
| " | " | 1 | " | CO, | 1.254 " |
| " | " | 1 | " | N, | 1.256 " |

Multiplying the per cent. by the volume of each gas by the weight of a liter of that gas, we get certain values, a, a', a'' , etc. Taking the sum of these = s , then the per cent. of weight would be $\frac{a}{s}, \frac{a'}{s}, \frac{a''}{s}$, etc.

To get the total O and the total C: The atomic weight of O = 16 and of C = 12; \therefore the amount of O in $\text{CO}_2 = \frac{2 \times 16}{12 + (2 \times 16)}$
 $= \frac{8}{11}$ and the amount of C = $\frac{12}{12 + (2 \times 16)} = \frac{12}{44} = \frac{3}{11}$.

In the same manner the amount of O in CO = $\frac{16}{12 + 16} = \frac{4}{7}$,
 and C = $\frac{3}{7}$.

Hence the total O = $\frac{8}{11} \text{CO}_2 + \frac{4}{7} \text{CO} + \text{O}$, and the total C
 $= \frac{3}{11} \text{CO}_2 + \frac{3}{7} \text{CO}$.

To get the ratio of air for dilution to that for combustion,
 we have $\frac{\text{Free O}}{\text{Combined O}} = \frac{\text{O}}{\frac{8}{11} \text{CO}_2 + \frac{4}{7} \text{CO}}$.

To get a measure of the air supplied per pound of carbon, we take the per cent. by weight of total O + per cent. by weight of N, and \div by the per cent. by weight of C.

At the time of taking the samples of gas the conditions were as follows:

No. 1. Fire had not completely burned clear from first firing. Back damper was wide open, as were the draught doors.

No. 2. Fire burning clear. Back damper dropped 3 inches. Draught doors wide open.

No. 3. Fire clear. Back damper 3 inches down. One draught door closed.

No. 4. Fires clear. Back damper 6 inches down. One draught door closed.

No. 5. Same as No. 4.

From these figures the following results are obtained :

FLUE GASES.

| | | |
|-----------------------|------------|-----------------|
| Average free O, | by weight, | 7.108 per cent. |
| " " CO ₂ , | " " | 17.059 " |
| " " CO, | " " | 2.584 " |
| " " N, | " " | 73.34 " |

The average ratio of the amount of air for *dilution* of the gaseous products of combustion to that necessary for combustion is as 0.514 to 1, *i.e.*, 16.44 lb. of air per pound of combustible, or 1.37 times the theoretical amount. The ratio of amount of air required for the dilution of the gaseous products of combustion to that necessary for combustion is variously estimated by different authors, but is generally taken as $\frac{1}{2}$:1. It will be seen that a very small per cent. of CO passed up the chimney, the average being 2.67 per cent. by volume, showing the combustion to be nearly complete.

The waste by air in the chimney is calculated by the following formula :

Let W = the number of pounds of air for combustion and dilution ;

t = temperature of chimney ;

t' = temperature of external air ;

S = specific heat of air.

Then

$$H = W(t - t')S,$$

where H is the number of heat-units carried off by the escaping gases.

We have

$$\begin{aligned} W &= 16.44; \\ t &= 435.7^{\circ} \text{ Fahr.}; \\ t' &= 60.39^{\circ} \text{ " } \\ S &= 0.238. \end{aligned}$$

Hence

$$H = 16.44 (435.7 - 60.39) 0.238 = 1468.48 \text{ units.}$$

Assuming that a pound of coal will evaporate 15 pounds of water from and at 212° Fahr., or equal to 14,491 heat-units, the loss by chimney is 0.101.

The height of chimney required under the above conditions is found from Rankine's formulæ as follows:

- Let W = weight of fuel burned in the furnace, per second;
 V_0 = the volume at 32° F. of air supplied per lb. of fuel;
 T = the absolute temperature of gas discharged by the chimney;
 A = sectional area of damper opening.

Then the velocity of the current in the chimney in feet per second is

$$u = \frac{WV_0T}{AT_0}.$$

Hence

$$u = \frac{0.06869 (12.386 \times 16.45) 896.9}{2.222 \times 493} = 11.449 \text{ ft. per sec.,}$$

and h , the head required to produce this draught, is

$$h = \frac{u^2}{2g} \left(1 + G + \frac{fl}{m} \right),$$

where

l = the whole length of chimney and flue leading to it in feet ;

m = its hydraulic mean radius ;

f = coefficient of friction ; (estimated by Pectet at 0.012 ;)

G = a factor of resistance for the passage of air through grate and fuel ; (given by Pectet as 12.)

Hence

$$h = \frac{(11.449)^2}{2 \times 32.2} \left(13 + \frac{0.012 \times 93}{534} \right) = 30.7138 \text{ ft.}$$

Then

$$H = h \div \left(0.96 \frac{T_1}{T_2} - 1 \right),$$

where H is the height of chimney :

$$H = 30.7138 \div \left(0.96 \frac{896.9}{521.59} - 1 \right) = 47.25 \text{ ft.}$$

The actual height as measured was 60.25 ft. The difference between this and the calculated height, or the throttling effect of the damper, being

$$60.25 - 47.25 = 13 \text{ ft.}$$

The following data were taken during the trial :

| | |
|------------------------------------|-------------|
| Total coal, | 2963.2 lbs. |
| Total ash and waste, | 342 " |
| Per cent. ash and waste, | 11.5 " |

The wood used was considered as equal to 0.4 the same weight of coal.

At 6 P.M. the fire was hauled and the unconsumed coal and the contents of the ash-pit were weighed up dry. The height of water in the gauge-glass was brought to the same position

as at the start, and all conditions made as near those at the beginning of the trial as possible.

The following are the records :

Total weight of water, . . . 23, 912.5 lbs.
Average temperature, . . . 46.125 Fahr.

AVERAGE PRESSURES.

Mercury-gauge, 85.78 lbs.
Edson gauge, by record chart, corrected, 85.4 "

Barometer readings were taken from the report of the University Signal Service.

Let x = weight of dry steam run into calorimeter ;

y' = weight of water in the steam ;

y = percentage of priming ;

W = weight of condensing water ;

w = weight of condensed steam ;

t'' = the initial temperature ;

t' = the final temperature ;

T = heat-units per lb. of steam ;

t = heat-units per lb. of water.

Then

Range of temperature, . . . $R = t' - t''$;

Heat transferred to calorimeter, $U = WxR$;

Heat from steam, per lb., . . $H = T - t'$;

Heat from water, per lb., . . $h = t - t'$.

$$x + y' = w, \quad \dots \dots \dots (1)$$

$$Hx + hy' = U. \quad \dots \dots \dots (2)$$

From 1 and 2,

$$x(H - h) = U - wh,$$

$$x = \frac{U - wh}{H - h}.$$

Percentage of priming,

$$y = 100 \frac{w - x}{w}.$$

The ten calorimeter experiments gave the following average results :

| | |
|-----------------------------------------------|--------------|
| Steam-pressure, | 100.522 lbs. |
| Weight of condensing water, | 382.985 " |
| Weight of steam condensed, | 24.335 " |
| Initial temperature, | 47.979 Fahr. |
| Final temperature, | 117.430 " |
| Range of temperature, | 69.45 " |
| Dry steam run into the calorimeter, | 24.2853 lbs. |
| Per cent. of priming, | 0.189 |

DATA AND RESULTS.

| | |
|------------------------------------------------------------------|----------------|
| Date of test, | April 28, 1887 |
| Weight of wood used in lighting fires, | 245.5 lbs. |
| Equivalent value of wood referred to fuel, | 98.2 " |
| Weight of anthracite coal used, | 3265 " |
| Total weight of fuel, | 3363.2 " |
| Weight of unconsumed coal left on the grates, | 400 " |
| Total weight of fuel consumed, | 2963.2 " |
| Weight of ashes and clinkers, | 342 " |
| Percentage of ash and clinkers to fuel consumed, | 11.5 |
| Percentage of moisture in coal, | 3.81 |
| Weight of fuel, less moisture, | 2752.11 " |
| Weight of combustible used, | 2410.11 " |
| Total weight of feed-water supplied and evaporated, | 23912.5 " |
| Average steam-pressure, | 85.4 " |
| Average temperature feed-water, | 46.71 Fr. |
| Average temperature of escaping gases, | 435.7 " |
| Average force of draught in inches of water, | 0.275 in. |
| Water evaporated per lb. of fuel, observed conditions, | 8.68 lbs. |

| | |
|-------------------------------------------------------------------------------------------------------------------------------|-------------|
| Equivalent evaporation, per lb. of fuel, from and at 212° Fahr., | 10.486 lbs. |
| Water evaporated per lb. of combustible, . . . | 9.92 " |
| Equivalent from and at 212°, | 11.984 " |
| Average temperature of boiler-room, | 80.06 Fr. |
| Average temperature of outside air, | 60.39 " |
| Average height of barometer, | 28.702 in. |
| Horse-power developed on a basis of 30 lbs. of feed-water supplied at 100° Fahr. and evap- orated at 70 lbs., | 83.75 |
| Rated horse-power, | 61 |
| Per cent above rated capacity, | 37 |

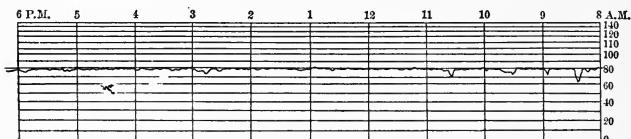


FIG. 11.—Autographic record of steam-pressure during the trial, from Edson gauge.
Mean pressure as shown on the diagram, 78.4 lbs. per sq. in.
Mean pressure, corrected, 85.4 " " "

CHAPTER IV.

THE STEAM-ENGINE INDICATOR.

50. The Indicator and the Dynamometer are the instruments employed in the engine-test proper. The purpose of their use is the measurement by the one of all fluctuations of pressure and of volume of the steam within the working cylinder, and of the work done and power developed by its action on the piston, the gross work performed by the transformation of heat-energy, and by the other the net work of the engine, the work done and power available at the engine-shaft for useful application. The difference between these two quantities is the measure of the lost energy and the wasted power, due to the resistances of the machine itself, the sum of the friction-resistances and the back pressure on the exhaust side of the piston, if the gross indicated power is measured to the line of external atmospheric or condenser pressure, or to friction alone if the power is taken as exclusive of back-pressure work.

The indicator is sometimes a "continuous indicator," giving a running, continuous, record of power developed. The most usual form, however, is that which gives a graphical representation of the whole cycle on one side of the piston, and thus permits a study to be made of all the variations of pressure throughout the stroke, and thus a deduction of the conditions of valve adjustment or setting, and of its action in distributing steam. The dynamometer is sometimes of the transmitting form, stationed between the engine and its work or any introduced resistance; but it is most usually of the type known as the Prony brake or the absorbing dynamometer, and takes up the whole external power of the engine, converting

all that energy into heat; which heat it wastes by conduction and radiation to surrounding objects or to a stream of water kept flowing over it, or through the rim of the brake-wheel.

51. The Principles of the Indicator of the usual construction and type, those which govern its action and determine its value, are as follows:

(1) It must exhibit with precision the pressure of the steam within the working cylinder at every instant throughout the stroke.

(2) Simultaneous measures must be given of the position of the piston corresponding to the given pressure, each instant.

(3) The diagram produced must be so made, automatically, as to have its ordinates exactly proportional to the steam-pressures and its abscissas as accurately proportional to the motions of the piston, each point in the curve, by its coördinates, giving a measure, simultaneously, of these two quantities.

(4) The diagram must be unaffected either by the forces acting on the engine, other than that which it is constructed to measure, or those brought into existence by its own motions, and whether they are active or passive, whether of inertia or of friction. The ideal indicator would be an instrument possessing the above qualities, and would trace a conveniently large diagram with absolute exactness. It would be free from inertia and perfectly inflexible in every part. As these ideal conditions are approximated, differences among the best makes of indicators become less and less, and should finally disappear. As they are now sometimes made, however, unless carefully selected and as carefully tested and standardized, it is perfectly possible for differences of very considerable importance to be observed. The Author has sometimes noted results from indicators, simultaneously used, varying from ten to fifteen per cent.

52. The Essentials of a Good Indicator are:

(1) Such form and construction as will insure its meeting the prescribed general conditions—accuracy of representation of the variations of steam-pressure and the simultaneous movement of the piston at all times.

(2) Such simplicity of form as will make it free from liability to accident and failure in operation.

(3) Such lightness of parts, and such rigidity as a whole, as will prevent any inaccuracy of indications arising from its inertia.

(4) It should be easily, conveniently, and safely attachable and removable, and readily and handily manipulated.

Stiffness, lightness, and exactness of standardization are the prime essentials. The springs should be exactly standard; the moving parts as light as is consistent with proper strength and stiffness; the stationary parts should be carefully proportioned and rigid; the whole instrument should be portable, and yet the scale of its diagram as large as practicable, and consistent with exactness in its production.

53. The Forms of the Indicator, as commonly constructed,

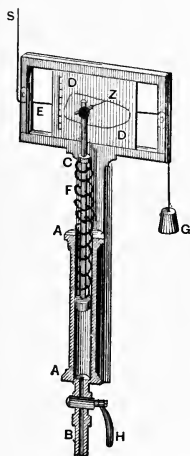


FIG. 12.—THE WATT INDICATOR.

constructed, are usually very similar, the more important differences being found in the recording system. The original indicator employed from about 1814* by Watt, Fig. 12, consisted of a small steam-cylinder, *AA*, traversed by a piston, *K*, the latter held by a spring, *F*, which was compressed or extended proportionally to the pressure, the cylinder being placed in communication with the interior of the working cylinder by a pipe, *B*, of sufficient size and fitted with a cock, *H*, by means of which the steam could be cut off from the instrument at any instant. So long as this cock was open, the indicator, if properly mounted, and the main steam piston were affected by precisely the same intensity of pressure, and the movement of the former was a measure of the pressure on the latter. A pencil, *z*, was attached to

* See Tredgold on the Steam-engine. London, 1827.

the indicator-piston, and its point recorded all such variations of steam-pressure on a movable slate, *D*, which was so connected, through *SE*, with the mechanism of the engine as to move in exact coincidence with the main piston, and precisely at right angles to the line of motion of the pencil. Thus, the abscisses of the curve produced were proportional to the motions of the piston, and the ordinates of the same points in the curve gave the simultaneous pressures.

In the later instruments of McNaught and Hopkinson, metal cylinders revolving on vertical axes were substituted for the sliding panel of Watt's arrangement, and a much more compact instrument was thus made.

McNaught's indicator, which was in general use until about 1860, when the first of the more modern forms, that of Richards, was introduced, had the form seen in the sketch, as described by Rankine, about the above date.*

AB is the barrel. Its lower end, *A*, contains a small cylinder, fitted with a piston, which cylinder, by means of the screwed nozzle at *A*, can be fixed in any convenient position on either end of the cylinder. The communication between the engine cylinder and the indicator cylinder is made by means of the cock, *K*.

The upper end, *B*, of the cylindrical case contains a coiled spring, one end of which is attached to the piston, and the other to the top of the casing. The piston is pressed from below by steam, and from above by the atmosphere. When the pressure of the steam exceeds that of the atmosphere, the piston is driven upward, and the spring compressed; when the pressure of the steam is less, the piston is driven downwards, and the spring extended.

A short arm *C*, a pointer *D*, which shows the pressure on a scale whose zero denotes the *pressure of the atmosphere*, and which is graduated upwards and downwards from that zero.

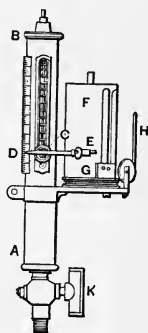


FIG. 13.—McNAUGHT'S INDICATOR.

* Steam-engine, p. 47 *et seq.*

At the other side, the short arm has a longer arm, carrying a pencil *E*. *F* is a brass paper-drum, which rotates backward and forward about a vertical axis, and which, when used, carries a piece of paper called a "card." The cord *H* is to be connected with the engine in any manner which shall insure that the velocity of rotation of the drum shall bear a constant ratio to that of the engine piston.

The later devices have been introduced with a view to securing lightness of parts and reduced motion of piston.

Fig. 14 is a sketch, partly in section, of the first of the

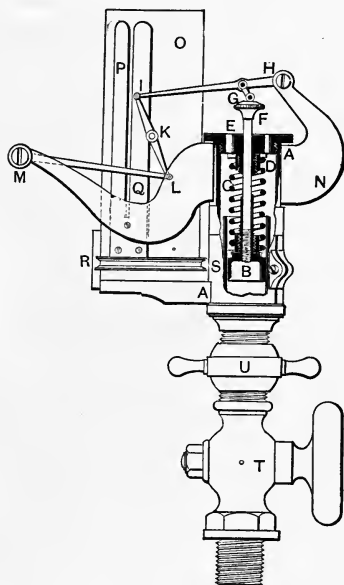


FIG. 14.—THE RICHARDS INDICATOR.

later type of instrument, the Richards indicator, invented by Professor C. B. Richards about 1860. *AA* is the cylinder; *B* is the piston, connected by a properly made spring, *CD*, with

the cap, *E*, of the barrel. The head of the piston-rod, *F*, is attached by a link, *G*, to the lever, *HI*, by means of which a comparatively large motion of the pencil, *K*, is obtained without much movement of the piston and its attached parts, and consequently with but little inertia-effect. A parallel motion of the Watt type, *HI, KLM*, guides the pencil-holder, *K*, in a right line parallel to this path of the piston of the indicator. The paper is wrapped about the cylinder, *O*, and secured at its ends by the clamps, *PQ*. The paper-cylinder is turned on its axis by a cord on the pulley, *RS*, which cord is attached to some form of "reducing motion" which causes it to move with the engine-piston.

Communication with the engine-cylinder is established by a steam passage through the cock, *T*, and the instrument is secured in place by the clamp *U*. When in action, this cock is opened; the indicator-piston rises and falls with the varying pressure in that end of the engine-cylinder, and the paper-barrel rotates backward and forward as the engine-piston moves. When all is ready, the instrument being heated up and working smoothly, the pencil is pushed lightly against the paper, and a diagram is drawn, representing all changes of pressure and volume of the working fluid during the period of contact. This modification of the indicator was found to give satisfactory results up to a comparatively high speed, and its limit of efficiency was determined by the degree to which the lightening of its parts could be safely carried.

A still later form (1875) is that of Mr. J. W. Thompson, Fig. 15. In this indicator the same general style is retained, but the parallel motion is modified. The

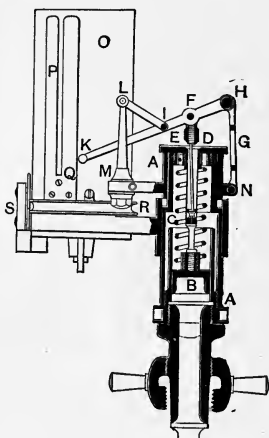


FIG. 15.—THE THOMPSON INDICATOR.

The

cylinder, *AA*, contains a piston, *B*, connected by a spring, as before, to the cap, *DE*; while the head, *F*, of the rod actuates a pencil-arm, *HK*, and a parallel motion is obtained by linking on *LI* from the standard, *LM*, and *G* from the swivelling support *MN*, which also carries *L*. The action of the instrument when in use is precisely as before; its decreased weights of moving parts, however, enabled it to be confidently relied upon at speeds far above those of even the Richards instrument. The old McNaught indicator became unsatisfactory at about 60 revolutions per minute; the Richards carried this limit well up toward and sometimes above 200 revolutions; while the Thompson indicator was found capable of doing good work on even the fast engines of the most modern type at the date of its invention.

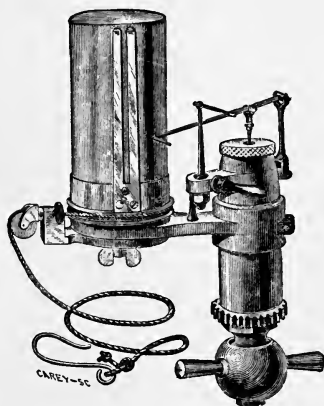


FIG. 16.—THE THOMPSON INDICATOR.

The most recent and a still lighter style of this instrument is shown in Fig. 16.

The later improvements consist in lightening the moving parts, substituting steel screws in place of taper pins, using a light steel link instead of a brass one, reducing the weight at the pencil-lever and elsewhere, shortening the length and reducing

the weight of the paper-cylinder one-half, and reducing friction to a minimum.

The paper-cylinder is so constructed that the tension of the coiled drum-spring within it can be varied for different speeds. As little or as much of the spring can be taken up or let out as desired, thus providing for fine adjustments.

Sufficient tension should be given to keep the cord taut at all points. When exceptionally accurate work is desired, the length of the diagram may be carefully measured, and compared with the length of a line traced on the paper when the engine is moved slowly. If the diagram is found to differ in length from this line, vary the tension of the spring till they agree.

All these indicators are provided with a piston 0.798 inch diameter, $\frac{1}{2}$ -inch area, and with springs for indicating pressures up to 250 pounds. When higher pressure is to be indicated, an extra piston 0.564 inch diameter, $\frac{1}{4}$ -inch area, is used, which, when substituted for the other piston, doubles the capacity of each spring, thereby adapting the indicator for indicating pressures up to 500 lbs.

The Tabor Indicator, Fig. 17, the invention of Mr. H. Tabor

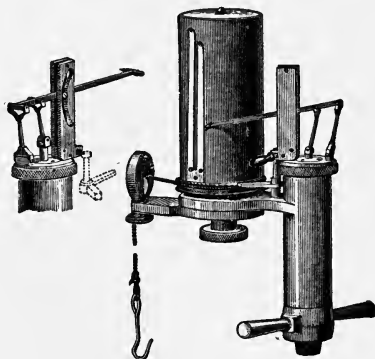


FIG. 17.—THE TABOR INDICATOR.

(1879), illustrates another ingenious attempt to evade all those difficulties incident to high speed of engines which have elim-

inated all the old forms of the indicator from the field. In this instrument the number of parts is still further reduced and the weight of such as remain is made as small as is thought safe. In the Tabor Indicator a stationary plate containing a curved slot is firmly secured in an upright position to the cover of the steam-cylinder. This slot serves as a guide and controls the motion of the pencil-bar. The side of the pencil-bar carries a roller which turns on a pin, and this is fitted so as to roll freely from end to end of the slot, with little lost motion. The curve of the slot is so adjusted and the pin attached to such a point that the end of the pencil-bar, which carries the pencil, moves up and down in a straight line, when the roller is moved from one end of the slot to the other. The curve of the slot just compensates the tendency of the pencil point to move in a circular arc, and a straight-line motion results. The outside of the curve is nearly a true circle with a radius of one inch.* The steam-cylinder and the base of the paper-drum are made in one casting. Inside the steam-cylinder is a movable lining cylinder, within which the piston of the indicator works. This cylinder is attached by means of a screw-thread at the bottom, and openings at the opposite sides at the top are provided for the introduction of a tool for screwing it in or out. Openings through the sides of the outer cylinder are provided to allow the steam which leaks by the piston to escape. The pencil mechanism is carried by the cover of the outside cylinder. The cover proper is stationary, but a nicely fitted swivel-plate which extends over nearly the whole of the cover, is provided, and to this plate the direct attachment of the pencil mechanism is made. By means of the swivel-plate, the pencil mechanism may be turned so as to bring the pencil into contact with the paper-drum, as is done in the act of taking a diagram.

The pencil mechanism is attached to the swivel by means of the vertical plate containing the slot, which has been referred to, and a small standard placed on the opposite side of the swivel for connecting the back link. The connection between the piston and the pencil mechanism is made by means

* "The Tabor Indicator;" G. H. Barrus, N. Y., 1888.

of a steel piston-rod. At the upper end where it passes through the cover, it is hollow and has an outside diameter measuring $\frac{3}{16}$ of an inch. At the lower end it is solid and its diameter is reduced. It connects with the piston through a ball-and-socket joint. A number of shallow grooves are cut upon the outside of the piston to serve as a so-called water-packing.

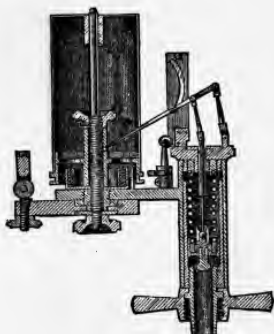


FIG. 18.—THE TABOR INDICATOR.

The springs used in the Tabor Indicator are of the duplex type, being made of two spiral coils of wire with fittings, as shown in the cut. The springs are so mounted that the points of connection of the two coils lie on opposite sides of the fitting.

The Crosby Indicator, Fig. 19, is still another successful recent type of the instrument (1879), and one which also illustrates that remarkable combination of lightness and accuracy which characterizes all good indicators. In this case, a still different form of parallel motion, light, stiff, and carefully adjusted, guides a very light pencil-holder carried at the end of a correspondingly light steel arm. The general arrangement of the indicator barrel and the paper-cylinder, with their attachments, is quite similar to those observed in the Richards and its successors.

If the conditions under which the spring acts be considered,

it is readily seen that, when the cord has the maximum other resistances to overcome, the drum-spring should offer minimum resistance. At the beginning of the stroke, when the spring is overcoming the inertia and friction of the drum, its resistance should be a maximum, and should gradually decrease. Here, a short spiral drum-spring is adopted, giving at the beginning of the stroke a comparatively slight resistance, which gradually

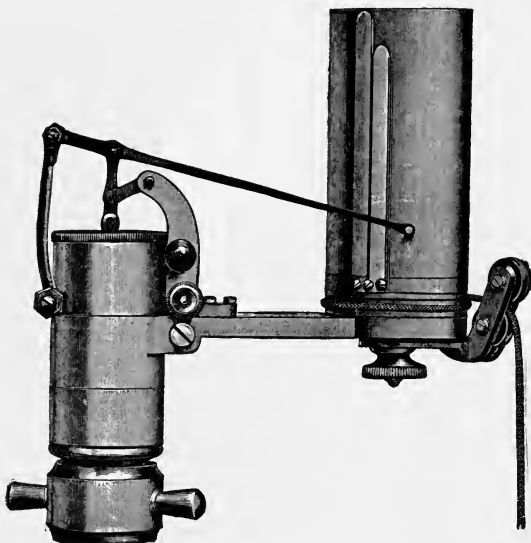


FIG. 19.—THE CROSBY INDICATOR.

increases until it reaches the maximum at the end of the stroke. In the other direction the recoil is strongest at the beginning of the stroke, and decreases to the end.

Duprez, Hirn, and Webb employ a screw, by means of which the steam is prevented from lifting the piston until the pressure exceeds a certain amount. Until this instant the indicator-diagram is a horizontal line; it then becomes curved. When

the screw is turned, the piston is again prevented from moving until the pressure exceeds a certain other limit, so that a series of corners is obtained, which are points on the real indicator-diagram, and may be joined by hand. When vibrations of the spring are in this way destroyed, exactly the same indication may often be obtained during four or five successive strokes.*

Mon. Hirn would prefer, where practicable, a directly connected spring, of considerable amplitude of range, stretching and compressing it by means of the screw just described, and allowing the attainment of the pressure registered at any instant to be indicated by the slight vibration or jump permitted by the lost motion in the grip on the spring as the steam-pressure passes that point. In these indicators it is evident that the

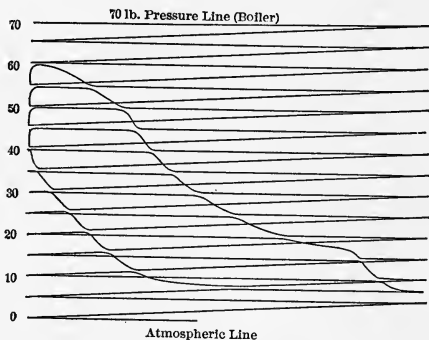


FIG. 20.—WEBB'S DIAGRAM.

diagram produced is then a "composite" of a number of successive indications, taken in as many successive revolutions of the engine. At the high speeds for which only such instruments are designed this is probably no disadvantage; but the instrument cannot show what occurs throughout any one revolution. The screw is usually so attached, however, that it may be readily removed at any moment, and the indicator thus quickly converted into the common form.

* Perry, *The Steam-Engine*, 1874; *Bulletin de la Soc. Ind. de Mulhouse*, 1876; *London Engineering*, Dec. 14. 1888, p. 576.

The indicator of Professor Webb is intended for use on fast-running engines where the inertia of the parts of the standard type of instruments is embarrassing.

Fig. 20 is a diagram taken at 400 revolutions per minute. The series of zigzag lines is the blank diagram. Each line is made by one stroke of the engine, and at 400 revolutions it would be about three seconds before the diagram is finished. If the indicator cock is open, the pencil will make the diagram. Instead, at the start, of following the 55-lb. line, it jumps up until the pressure on the cylinder falls again to 55 lbs., when it will come back to the line and finish it. The pencil will then return to the left side on the diagonal line, and the process will be repeated until the card is complete.

In Fig. 21 an indicator is shown with this device attached.* The frame of the instrument is extended up to *e*, and a hole is made through it in which the screw *b* slides freely. This screw

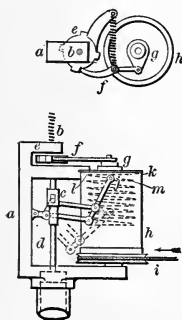


FIG. 21.—THE WEBB INDICATOR.

has a nut, *e*, which is confined between the forks of the frame. The lower end of the screw has a forked head, *c*, which embraces the upper end of the piston-rod, and is attached to the same by means of a pin, which passes through holes in both; the hole in the fork is, however, slotted out a little larger than the pin, as will be seen hereafter. If the nut, *e*, is revolved, the screw will draw up the piston against the spring contained in the indicator cylinder until we have set the latter at any desired pressure. If this pressure be above the maximum pressure in the engine cylinder, the pin at *c* will remain resting on

the bottom of the slot; but if it be below, then during any stroke of the engine it will remain there only so long as the cylinder pressure is below that of the spring; it will jump to the top of the slot when the former exceeds the latter. The slot is made the right length to allow of a jump of 4 or 5 lbs.,

* Trans. Am. Soc. M. E., vol. ii.

as shown. To complete the arrangement we add the pawl *f* and crank *g* mounted on the top of the indicator-drum, *h*, and so arranged that during each backward stroke *f* shall revolve *e*, and thus let the screw, *b*, with the piston and spring, gradually down.

This instrument thus produces the mean of several diagrams, making the whole by combining parts of each.

The indicator is sometimes given the form shown in the accompanying engraving, in order to obtain a record of the

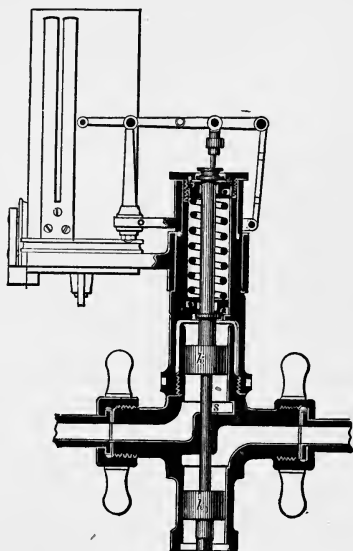


FIG. 22.—THE DOUBLE INDICATOR.

net pressure on the piston. One piston receives the steam-pressure at one end of the steam-cylinder; the other is acted upon by the steam on the other side of the piston. The effort on the spring is thus the net pressure, and the diagram produced, as shown in the next figure, presents this net pressure

at every instant throughout the stroke. The upper portion, *abkdl*, and the lower part, *fgedk*, measure the work done in the forward and backward strokes, respectively.

Every modern indicator thus combines the essential elements of construction which have been seen to be required to secure accuracy of diagram, and in a very admirable manner. The latest and best may be relied upon to give good "cards"

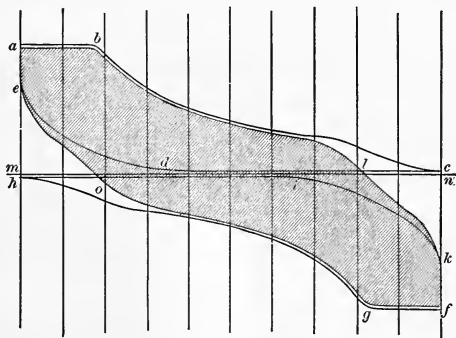


FIG. 22a.—DOUBLE DIAGRAM.

at any speeds yet attained in usual operation by the fastest of the "high-speed" engines.

54. The Standardization and Test of the Indicator should always precede its use. To give satisfactory results, the instrument must give a diagram of which the abscissas shall exactly represent the successive positions of the moving piston, while the ordinate of each as exactly measures the simultaneously occurring pressure within the working cylinder. The weight and inertia of every moving part of the indicator introduce errors which, while they may not be completely eliminated, may, by reduction of size and special expedients, at least be rendered so small as to be unimportant at any ordinary speeds. They are, however, more difficult of prevention as the speed of rotation and the pressures adopted increase. In a well-made instrument the spring will precisely measure pressures, the

pressure in the indicator will be sensibly the same as in the engine, and its piston will move so freely as not to affect the indications by its resistances due to close fitting. In all, however well made, on the other hand, it is found impossible, by any art of design or construction, to wholly eliminate the influences of inertia of moving parts, friction of joints and guides if the latter be used, and of piston or pencil, or the effects of variation of spring tension on the motion of paper-cylinder and card. Standardization is the process of detecting and measuring such errors as are observable in the action of the apparatus, and the determination of their influence on the indications obtained by its use.

Springs are tested most satisfactorily by connecting the indicator, with its appropriate spring in place, with a small steam-boiler or steam-reservoir or convenient steam-pipe, in such manner that simultaneous measurements of pressure may be taken by the indicator record and a standard test-gauge known to be correct. If the spring be tested cold, it will be found inaccurate if it had been found right when hot; and the correct reading of a cold spring is evidence that it is not right under steam;* since, when the indicator is in use and the spring heated, both by the steam leaking past the piston and by conduction through its attachments from the piston and from the indicator-barrel, its strength and its elasticity are sensibly modified, the spring being thus weakened. In thus testing springs, such arrangements should be made as will enable the observer to hold the pressure at any desired point until readings from the standard test-gauge can be deliberately and precisely taken. If the spring is found unreliable, it should be at once exchanged for a good one.

“Throttling,” or loss of pressure between the engine and the indicator, is produced by long, tortuous, or contracted passages. The connections should be as large, as straight, and as short as practicable, and, other things being equal, that indicator is best in which steam connections can be best effected.

* This difference has been found to amount to $2\frac{1}{2}$ or 3 per cent.—Proceedings of Brit. Inst. C. E., 1885; Brightmore; discussion.

The cock under the indicator should have an opening fully as large as the pipe itself. It should also have a hole bored in from one side for the purpose of freeing the instrument and connections from water of condensation coming over with the steam. The holes in the cock, or the upper part of the barrel of the indicator, should be large enough to permit free egress to any steam that may leak past the piston; and the latter is, in all good makes, so loose as to leak observably under pressure. The effect of leakage is insensible; but were the fit a tight one, the resulting friction might be important. It is to insure this exact correspondence of pressure in the working cylinder of the engine and in that of the indicator that it is customary, on all high-speed engines, to employ indicators simultaneously at both ends, thus obtaining very short and direct steam connections.

The springs should be so made and fitted that their action under pressure, and when in use, may not throw the piston out of line or cramp it in the barrel, and thus produce what are sometimes found to be serious errors. A small leak does no harm, and is, on the whole, desirable. The piston should move so freely that, the spring being removed, the breath may blow it from end to end of its barrel and draw it back again.

The friction of the pencil on the paper is probably closely proportional to the force with which it is pressed upon the latter, and variable with the texture of the paper and the sharpness of the pencil-point, and its material. This is often an important source of error in the diagram. The reacting effect of this pressure on the pencil mechanism is also, but in comparatively slight degree, a source of inaccuracy of record. The result of such frictions is the production of an enlargement of the "card" to the extent, often, of a very appreciable, and sometimes of an important, amount. This friction is sometimes relied upon to diminish those oscillations of the instrument which at high speeds render the diagram difficult of measurement, or even untranslatable. In such cases the real power of the engine may be several per cent. less than shown by the in-

strument. To avoid this difficulty, it is best to make the pencil bear on the paper only just hard enough to make a visible mark. A hard-lead pencil or one of soft metal smoothly pointed, paper having a "metallic" or glazed surface, and a light, steady pressure producing an extremely fine but perfectly visible line are the conditions to be sought. In such case, the error due to friction of pencil will be inappreciable.

The stretching of the cord turning the paper-barrel is a common cause of inaccuracy in length and of distortion of the diagram. The varying tension of the spring, and the surges due to the inertia of the rotating mass, together cause variations of length of the cord that may give rise to errors of really important magnitude. The string, even when its primitive stretch is taken out of it by a preliminary application of a heavy load, retains some elasticity, and will have a sensibly variable length under the constantly varying pull when in use. Any observable friction of the paper-barrel also tends to exaggerate this action. The inertia of the drum tends to compensate this effect, and it is possible to so adjust the strength of the spring to this inertia as to make the variation of stress on the cord comparatively small.

The effect of this stretch is to cut off a part of the diagram at one end, and it is perfectly possible thus to reduce the apparent indicated power of the engine 10 or even 20 per cent. below the correct quantity.* The longer the cord, the greater the error; and differences of sensible amount may often be detected between the diagrams from opposite ends of the same cylinder, in area of diagram and in point of cut-off and amount of expansion, produced by differences in length of the cord used. The higher the speed of rotation of the engine, the greater the amount of this error; and instruments giving perfectly satisfactory cards at low speed may produce very defective diagrams at high speed. The lighter the drum and the spring found practicable, the better the results. A cord should always be well stretched before use; but a fine steel wire

* Proc. Brit. Inst. C. E., 1885: Brightmore.

is much to be preferred.* The paper-cylinder or drum should have carefully adjusted springs for fast work. The difference in its initial and final tension should be as nearly as possible equal to the inertia-stresses of the drum. Improvement has been carried so far in the reduction of weight of paper-barrel as to bring it, in one case at least, as low as 209 grammes (3088 grains). It has been often proposed to use aluminium for moving parts in order to reduce inertia-effects to a minimum. Errors due to this action, in good instruments, fall much below 1 per cent. In adjusting the instrument, the tension of the drum-spring should vary as the square of the number of revolutions; and it should be set for any speed in such manner that the length of the diagram should be the same at starting as when at full speed, as nearly as possible.

In testing the action of the drum-spring, either of two or three devices now in use may be employed. The accompanying sketch shows that devised by Mr. Brown.



FIG. 23.—DRUM-TESTING APPARATUS.

This instrument was designed to show the strains on the cord. From its diagram may be calculated the errors due to the stretch of the cord. The testing instrument consists of a plate, *A*, to one end of which is fastened the frame, *BB*, carrying the slide, *C*, and its cross-head, *D*. The spring, *F*, is screwed to the cross-head; the other end is connected with the lever, *G*, carrying the pencil. The rod, *E*, which moves the slide, *C*, receives its motion from a crank not shown. The swinging leaf, *F*, holds the paper on which the diagram is to be taken. The indicator is clamped to the plate, and the drum-cord connected with the spring. The crank is

* Mr. Wallace finds the yield of a good ordinary stretched indicator-cord to be from 0.008 to 0.0125 per foot per pound; and of wire No. 36 B. W. G. 0.003.

made to move at the speed desired. The paper is then raised to the pencil, and the diagram taken. If the strain on the cord is constant, the forward and return strokes will be parallel; but if the strain is not constant, the pencil will rise and fall as the strain varies. The line below the diagram is the line of no stress, drawn when the cord has been detached from the indicator.

The diagrams are shown two thirds their original size.

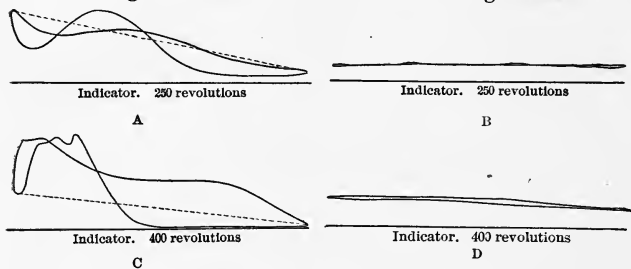


FIG. 24.—DRUM-STRESS DIAGRAMS.

Oscillations of the pencil about its proper position, and the consequent production of wavy lines in the diagram, cause the most serious defects in diagrams taken at high speeds of engine. Such deformations of the diagram are due to the inertia of the pencil and its holder and connections, and become greater as speeds increase, until, with every instrument, a speed is finally reached at which the diagram becomes unintelligible, as in the figure on page 148, which represents the card obtained by using an old style of indicator, with a light spring, at 300 revolutions per minute. With the best modern indicators it is easy to secure a perfectly smooth diagram at this speed.

These vibrations are the more serious as the proportion of the weight of the moving parts of the indicator are the heavier and as the spring is lighter. Their effect is not only to disguise the true form of the diagram, but also to enlarge it and thus to give too great values of power developed. Professor Reynolds gives the following as speeds at which this variation becomes one per cent. in an indicator having a piston-area of one-half

square inch and a weight of moving parts equivalent to 0.33 pound at the piston : *

| Spring used, lbs. per sq. in. | No. revolutions per minute. |
|----------------------------------|--------------------------------|
| 20 | 166 |
| 40 | 237 |
| 60 | 288 |
| 80 | 332 |
| 100 | 371 |

This error varies directly as the weight of the moving parts. In modern indicators of the best forms, it is probably inap-



FIG. 25.—INDICATOR VIBRATIONS.

preciable at all familiar engine speeds, the maximum being taken at about 300 revolutions per minute, five per second; and it may be assumed by the engineer that if his indicator is of good make, if he finds its parts correctly made, and if he keeps them in good order, he may rely, in all ordinary cases, on obtaining diagrams correct to within the limits of his nicest measurement, if the instrument is properly attached to the engine and skilfully handled.†

For higher speeds than are now obtained, indicators of the class illustrated by that of Duprez must be employed.

* Proceedings Inst. C. E., vol. lxxxiii., 1885.

† Barrus on Modern Indicators, and Discussion thereon : Trans. Am. Soc. M. E., vol. v. pp. 310-339. 1884.

The following is a description of Professor Reynolds' device for checking the movement of the drum, and to ascertain what distortion is caused by its irregular fling and friction : *

A Grove battery of five cells, in conjunction with a Ruhmkorff coil, is used. The wire from one pole was connected with one of the binding-screws (*H*) of the coil as usual, but the wire from the other pole of the battery was connected with the engine. A wire from the other binding-screw (*G*) was attached to the contact-breaker (*B*), a smooth piece of wood, into which pieces of wire were inserted at equal distances, the distance between the first and last wire being the length of the stroke of the engine. This was fixed on the lower slide, so that a pointer (*A*), secured to the cross-head, should slide on it. One

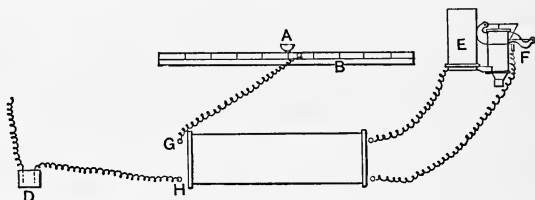


FIG. 26.—DISTORTION BY STRETCH OF INDICATOR-CORD.

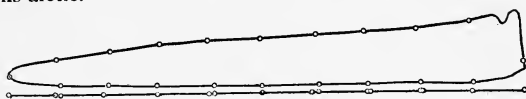
wire of the secondary coil was connected with the drum (*E*), and the other to a cup of mercury, into which the metallic pencil (*F*) dipped, thus completing the circuit when the pencil touched the paper.

In the following diagrams the relative positions of the circles show which parts of the diagrams are lengthened, and which are shortened. The effect is not merely to shorten the ends and lengthen the middle of the diagrams, but also to distort them, *i.e.*, to cause corresponding points not to lie in the same vertical line. The amount of this distortion is shown by the distance between corresponding points on the atmospheric line. (Fig. 27.)

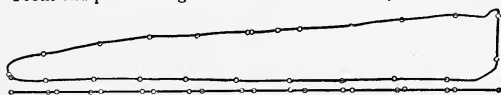
It is evident that the pencil may be made barely to touch

* Proc. Brit. Inst. C. E., 1885, No. 2070.

the paper without marking it and the diagram made by the sparks alone.



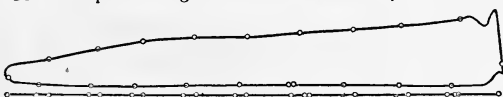
Front-end pricked diagram taken with wire at 107 revolutions.



Front-end pricked diagram taken with string at 107 revolutions.



Front-end pricked diagram taken with wire at 127 revolutions.



Front-end pricked diagram taken with string at 127 revolutions.

FIG. 27.—ELECTRIC DIAGRAMS.

Comparisons of indicators will often eliminate uncertainty as to their reliability. If an indicator is known to be right, the diagrams produced by it should be, under similar conditions, duplicated by an instrument the accuracy of which is doubted. Where three or more instruments are compared, the presumption is usually a fair one that, if one differs in any important degree where the others agree, it is defective. Where several are to be compared, it is sometimes practicable to take diagrams from them all simultaneously by fitting up properly. In such cases, all should be equidistant from the steam-cylinder and should have equally straight and large connecting pipes. One large pipe and cock, taking steam from the cylinder, terminated by the several pipes leading radially from its top to the several instruments, will usually answer the purpose.

In comparing the details of construction, it is to be remembered that the points to be studied are the exactness and per-

fection of dimensions and workmanship, and the weight of parts and their action as affected by inertia. The latter point has been seen to be peculiarly important when the instrument is to be used on engines at speeds exceeding about a hundred revolutions per minute.

Comparing, in this particular, the action of the paper-cylinders, or -drums, we find the time of a vibration, the forces freely acting, to be

$$t = 2\pi \sqrt{\frac{d}{a}};$$

in which d is the angular displacement, here to be reckoned from the position of mid-throw, and a the angular velocity. Then

$$a = \frac{4\pi^2 d}{t^2};$$

and

$$d = \frac{1}{2} \frac{\text{length diagram}}{\text{radius of drum}} = \frac{l}{r},$$

and the couple acting to start the drum into harmonic motion is

$$aI = \frac{4\pi^2}{t^2} I \times l.$$

That instrument, therefore, which has the least value of I the moment of inertia of the drum, is least liable to inaccuracy from this source of stress. The best adjustment should be sought in each case, and the comparison effected after this adjustment has been made. Since the effort of the drum-spring is usually directly proportional to the angle of motion, and since the force due acceleration is zero at mid-throw, if the difference of tension on the cord at the beginning and end of the motion is twice the effort required to overcome the inertia-resistance of the drum at starting, the action on the cord will be uniform when at speed, and the diagram entirely free from distortion from this cause. Other things being equal, that is

the best instrument in which this adjustment is secured. In some cases they are arranged for such adjustment at several usual speeds, or so as to permit the tension of the spring to be altered as desired—increasing it at high speeds, diminishing it at lower speeds. The acceleration for a $4\frac{1}{2}$ -inch card is $\frac{0.11523}{t^2}$, or for a 3-inch card $\frac{0.7703}{t^2}$; and, for the power, a pound acting at the circumference of a drum 2 inches in diameter will give the following maximum resistance at the stated speeds : *

| Revs. per min. | R. | Revs. per min. | R. |
|----------------|-----|----------------|------|
| 120 | 0.5 | 360 | 4.2 |
| 180 | 1.0 | 480 | 7.4 |
| 240 | 1.8 | 600 | 11.5 |
| 300 | 2.9 | 1000 | 32.0 |

These stresses evidently become serious at high velocities, increasing, as they do, as the square of the speed; and the higher the speed the greater the difference in favor of that instrument having lightest drum.

The stretch of the cord used (if sensible) should be observed; as this is an element which determines also the amount of distortion of the diagram due to the inertia of the drum and the action of its spring. In all cases we have the moment of the pull, P , on the cord,

$$Pr = \frac{d^2\alpha}{dt^2} + Rr + Fr;$$

in which the angular acceleration is $\frac{d^2\alpha}{dt^2}$, and Rr and Fr are the spring and the friction moments. The first quantity has been seen to be zero at mid-throw, its value increasing each way; the friction moment may be taken constant and unimportant, and the spring-resistance variable with its flexure, as already seen. The stronger and the less elastic the cord, and the better the adjustment of the spring-action to the inertia-effect, the

* Wallace on the Indicator : Trans. Inst. Scotland, 1888, p. 3.

more accurate the diagram. In good examples these effects are unimportant.

Comparing the indicators as to the effect of surges and oscillations, it is found that both these actions may become serious at high speed and with heavy pencils, springs, and pistons. The surge of the moving parts due to their rise or fall through the height of the diagram tends to increase the area of the curve. If indicators give similar and correct results at moderate speeds, this increase at higher speeds may be compared to determine their relative merits. It should never equal 1 per cent. This limit is found by Professor Reynolds, for the Richards indicator, at the speeds already given. It is seen that the maximum speeds of rotation in revolutions per minute is about

$$R = 40 \sqrt{s},$$

where s is the scale of the spring in pounds to the square inch; and this disturbance varies directly as the weights. This comparison may therefore be effected by weighing the moving parts.

The vibratory disturbance of the pencil is due to the elasticity of the spring, and its time is

$$t = 2\pi \sqrt{\frac{wr^2}{12pmg}};$$

where wr^2 is the effect of the moving parts reduced to the working-point; p , m , and g are the total load on the piston, the ratio of pencil and piston motion, and the acceleration of gravity. These disturbances are not serious as affecting the area of the diagram, but they are sometimes important as obscuring its meaning. A comparison of indicators in this regard would be made by taking diagrams at continually increasing speeds and noting the point at which the outlines of the figure become wavy and when they interfere with its legibility. It is seen that this defect increases as the square root of weight of moving parts, and is the more serious as the weights are nearer the

pencil and subject to rapid movements and quick changes of direction of motion. These irregularities may be partially controlled by the pressure and friction of the pencil, but only at the sacrifice of accuracy. The value of t should always be as small as practicable. If too large, the diagram may be seriously distorted. Any number of oscillations in the tracing of the card exceeding 25 or 30 may be permitted; less than 20 or 25 is objectionable.

Comparing springs, it will be often found that considerable differences are observable in their indications, both cold and hot. They should be examined to see that they take no permanent set, that they yield in exact proportion to the pressure, and that their attachment to the instrument is such as not to produce lateral strain or friction. Springs which have been already repeatedly given their full set by the makers are best. They should always be tested and compared hot.*

The best indicator, as is now evident, is that which, by such comparison and examination as has been described, is found to give the most exact and reliable diagram, and to be least affected by inertia-forces and the action of its own parts at high speed; it is that, in detail, which has proportionally the largest and lightest piston, the stiffest and lightest springs, the least friction of moving parts, the most perfect pencil mechanism, the most accurate and constant scale of pressures, the most perfect adjustment of drum-spring, and the lightest moving parts generally.

The following is the method of comparing indicators adopted by the Navy Department: †

A horizontal pipe, 2 inches diameter and 24 inches long, fitted with suitable pipes and valves for the admission and discharge of steam and provided with three nipples, two for the attachment of indicator and one for a steam-gauge, was used in the tests.

* See the valuable papers of Dr. Berndt on this subject in the Sächsische Ingenieur und Architekten Verein, 1882-85 meetings; and Lond. Engineering, 1877-8.

† Report of Chief of Bureau of Steam Engineering, 1888.

The steam-gauge having been secured in place, steam was blown through the test-pipe and indicator-nipples several times to free the pipe of water and dirt.

The indicators, after being well oiled, were secured in position and steam admitted, the pressure being allowed to rise until the limit for which the springs were designed was reached, in order to bring the instruments to their working temperature.

After trying the instruments to see that their movements were free, steam was discharged from the test-pipe and the indicator-cocks closed. The piston of each instrument was then pressed down slightly by hand and allowed to return to its normal position, with the friction of the moving parts opposed to the movement of the spring. When this had been done, the atmospheric line was drawn across the card. Steam was then admitted to the instruments and so regulated that the hand of the steam-gauge would rise slowly to the interval of pressure to be noted; and when it reached that point, at the word "mark," an operator stationed at each instrument drew the required line of its scale. All lines of the scales were drawn in the same manner, the top steam-line of the first test of each series being extended across the card.

Before beginning the down scales, the steam was allowed to rise a pound or two above the pressure to be first noted, in order to oppose the friction of the instrument to the movement of the spring. At the end of the down scale, the steam was shut off and discharged from the pipe and the indicator-cock closed before drawing the atmospheric line.

To determine the comparative indications of identically the same power by the two instruments, the following method was used :

The indicator-pipe at the outer end of the engine was fitted with a T and two right-angle branch pipes of equal diameters and lengths terminating in nipples. To the latter, the indicators were attached, after clearing the pipes of water and dirt and lubricating the cylinders. The springs of the paper-drums were adjusted to approximately the same tension. The cords

around these drums were tied to each other and to a single cord connecting with the indicator motion. This arrangement gave coincident motion to both drums without sensibly affecting the lead of the cords, as the angle between the latter was small.

One operator could readily take cards from both indicators at the same time.

Ten cards having been taken from each indicator, the latter were interchanged and then ten more cards taken from each. This change was made in order to eliminate any errors due to possible differences in the bore and lead of the branch pipes.

The test to determine the pencil movement was made as follows :

The spring of each indicator having been removed, a micrometer gauge was fitted to the cylinder and the weight of the piston and attachments taken on the end of the micrometer screw, the zero of the wheel coinciding with that of the vernier. A line was then drawn with the pencil of the instrument and formed the first one of the scale. The micrometer screw was then turned one revolution and a second line drawn ; this was repeated until the scale was complete for the movement of the piston.

A test to determine the line of motion of the pencil in each instrument was made as follows :

The spring having been removed, the piston was pushed up its entire stroke, the pencil at the same time drawing a line on the card, while the paper-drum was securely held by the detent attachment. Ten such lines were drawn on the card with each instrument.

All moving parts of both instruments were carefully weighed and their weights in Troy grains found.

At the conclusion of the foregoing tests, the steam-gauges used in the work were carefully compared with the mercury column.

The test of indicators, taking simultaneous diagrams from the same end of the engine, is liable to give misleading results unless great care is taken to have both equally well fitted and

similarly situated. To insure perfect fairness, they should be transposed and again compared. The following is the result of a comparison so made. The results can of course only be taken as gauging the work of the individual indicators so compared :

TESTS OF INDICATORS.

| Simultaneous cards taken from engine with A and B indicators fitted with 20-pound springs. | | | | | | | Simultaneous cards taken from engine with A and B indicators fitted with 40-pound springs. | | | | | | |
|--------------------------------------------------------------------------------------------|----------------|---------|-------------|------------------------|---------|-------------|--------------------------------------------------------------------------------------------|------------------------------------------------------------------|---------|-------------|------------------------|---------|-------------|
| Number of Cards. | Mean pressure. | | | Indicated horse-power. | | | | Mean pressure. | | | Indicated horse-power. | | |
| | A | B | Difference. | A | B | Difference. | | A | B | Difference. | A | B | Difference. |
| 1 | 13.30 | 12.65 | .65 | 26.797 | 25.487 | 1.310 | | 14.60 | 14.30 | .30 | 29.842 | 29.229 | .613 |
| 2 | 13.80 | 13.20 | .60 | 27.803 | 26.595 | 1.208 | | 14.80 | 14.00 | .80 | 30.251 | 28.616 | 1.635 |
| 3 | 13.65 | 12.85 | .80 | 27.502 | 25.890 | 1.612 | | 13.90 | 13.20 | .70 | 28.412 | 26.981 | 1.431 |
| 4 | 13.45 | 12.65 | .80 | 27.099 | 25.487 | 1.612 | | 13.50 | 12.75 | .75 | 27.594 | 26.061 | 1.533 |
| 5 | 13.30 | 12.55 | .75 | 26.797 | 25.286 | 1.511 | | 14.60 | 14.35 | .25 | 29.643 | 29.122 | .521 |
| 6 | 13.47 | 12.60 | .87 | 26.942 | 25.203 | 1.739 | | 13.00 | 12.45 | .55 | 26.572 | 25.448 | 1.124 |
| 7 | 13.62 | 12.58 | 1.04 | 27.243 | 25.163 | 2.080 | | 12.15 | 11.85 | .30 | 24.835 | 24.221 | .614 |
| 8 | 14.20 | 13.25 | .95 | 28.403 | 26.503 | 1.900 | | 12.52 | 12.50 | .02 | 25.591 | 25.550 | .041 |
| 9 | 13.90 | 12.85 | 1.05 | 27.803 | 25.702 | 2.101 | | 12.70 | 12.20 | .50 | 26.144 | 25.115 | 1.019 |
| 10 | 14.30 | 13.30 | 1.00 | 28.603 | 26.603 | 2.000 | | 12.50 | 12.00 | .50 | 25.550 | 24.528 | 1.022 |
| 1 | 16.95 | 16.16 | .79 | 33.161 | 31.615 | 1.546 | | 15.70 | 15.10 | .60 | 33.008 | 31.746 | 1.262 |
| 2 | 15.80 | 14.60 | 1.20 | 31.372 | 28.990 | 2.382 | | 14.00 | 12.75 | 1.25 | 29.025 | 26.433 | 2.592 |
| 3 | 15.50 | 14.28 | 1.22 | 30.664 | 28.250 | 2.414 | | 14.35 | 14.00 | .35 | 28.886 | 28.207 | .679 |
| 4 | 15.45 | 14.10 | 1.35 | 30.677 | 27.997 | 2.680 | | 17.00 | 16.43 | .57 | 33.259 | 32.144 | 1.115 |
| 5 | 17.00 | 15.65 | 1.35 | 34.003 | 31.303 | 2.700 | | 14.75 | 14.60 | .15 | 29.718 | 29.416 | .302 |
| 6 | 15.25 | 13.90 | 1.35 | 30.503 | 27.803 | 2.700 | | 18.40 | 17.25 | 1.15 | 36.535 | 34.252 | 2.283 |
| 7 | 14.20 | 13.00 | 1.20 | 28.403 | 26.003 | 2.400 | | 17.20 | 16.50 | .70 | 35.157 | 33.726 | 1.431 |
| 8 | 16.20 | 14.70 | 1.50 | 32.403 | 29.403 | 3.000 | | 16.20 | 15.70 | .50 | 33.113 | 32.091 | 1.022 |
| 9 | 15.20 | 13.80 | 1.40 | 30.403 | 27.603 | 2.800 | | 15.50 | 15.00 | .50 | 31.682 | 30.660 | 1.022 |
| 10 | 12.65 | 12.35 | .30 | 25.303 | 24.702 | .601 | | 14.85 | 14.50 | .35 | 30.353 | 29.638 | .715 |
| Means | 14.5595 | 13.5510 | 1.0085 | 29.0942 | 27.0794 | 2.0148 | | 14.6110 | 14.0715 | .5395 | 29.7585 | 28.6592 | 1.0993 |
| Power measured by A 7.4403 per cent. greater than measured by B. | | | | | | | | Power measured by A 3.8358 per cent. greater than measured by B. | | | | | |

NOTE.—First series: A on left-hand pipe. Second series: A on right-hand pipe.

This comparison has no value or meaning unless one instrument is known to be accurate and standard.

To test the friction of the working parts of the indicator, if a means can be secured of obtaining a manageable and variable steam-pressure, try the instrument at various pressures, as shown by a *reliable* steam-gauge, and compare the gauge-readings with those obtained by measurement of the diagram, as exhibited in the figures on page 158.

In *A*, Fig. 28, the diagram is that given by an instrument fitted with a "30-lb. spring," and having considerable friction of pencil movement; in *B*, the diagram is that of an indicator of little friction, and fitted with a "20-lb. spring."

How far such tests and comparisons may be taken as quantitatively gauging the value or accuracy of the instrument is

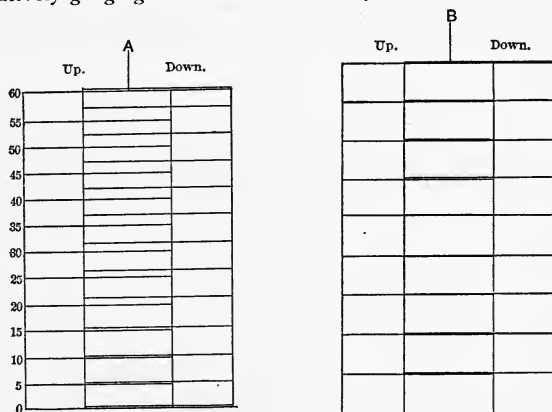


FIG. 28.—INDICATOR TEST.

uncertain. Probably the less the friction—as a rule, though not always—the better the indicator; but the action of movement in use, and the effect of inertia of parts, so modify final results, the former by lessening, the latter by exaggerating, the effect of friction, that it is quite impossible, so far as is to-day known, to predicate definite quantitative deductions relating to the accuracy of the instrument. We can only say that the lighter the parts, the less the friction, and the more accurate the spring-tensions and the pencil-movements, the better the indicator, and that the best now made, under the usual working conditions of the best engines, may be expected to give sensibly correct diagrams. This fact does not make it any the less imperative that every indicator to be used in any important work should be fully and carefully tested.

The standardization of the indicator is the more important from the fact that there is no available means of checking its work. The work of the engine, as otherwise customarily measured, is rarely known with accuracy, and the Author has known several indicators, used under similar conditions, to differ among themselves 10 and 15 per cent., with no means at hand of determining which of them were wrong, or the extent of their errors. Where a dynamometric brake is used, the check is more satisfactory, as the friction of the engine is commonly known, or ascertainable within a comparatively small limit of error. The best makers of indicators are, however, usually prepared to guarantee, to standardize, and to give variation-tables of their instruments; and the errors are now reduced in such cases to probably very small amounts.

55. The Attachment of the Indicator should always be so effected that its piston may receive precisely the pressure simultaneously acting on the engine-piston, and so that the motion of the paper shall exactly reproduce, as to time and in its proper proportion, the movement of the piston. This means that the steam-connection should be amply large and free from bends and angles, and that the cord and reducing motion giving movement to the paper-cylinder, or drum, should be so arranged as to lead right, and to be perfectly free from lost motion or stretch.

In attaching the instrument, it is usual to drill a half-inch hole in each end of the steam-cylinder, and to make connections with half-inch pipe to the indicator-cock as directly as possible. In many cases it will be found that the drilling has already been done by the builder of the engine. The opening into the cylinder is commonly in the clearance space back of the piston. Care should be taken that it is not covered by the piston at the end of stroke, and that the in-rush of steam from the steam-port is not likely to produce any sensible effect by blowing across the hole. Especial care should be taken to prevent chips from the drill falling into the cylinder and lodging where they can do injury. The work should, if practicable, be done with the heads removed; if this is not practicable, a

little steam should be turned on and the chips blown out before starting. If the indicator-cock can be screwed directly into the cylinder, it is an advantage. The indicator should, if possible, stand in the vertical position when in use, and one should be placed at each end of the cylinder, and diagrams taken as nearly simultaneously as possible. The cock between the indicator and the cylinder should be of the full size of the pipe, and should be so made that steam may be at any time either turned on the instrument or blown out into the air to clear the passages, and to see that all is right.

The Reducing Motion is made in many ways, and is often, by the ingenious engineer, improvised for the occasion. It must reduce the motion of the piston so as to give a correct throw at the drum and exactly proportionally at every part of the movement.

One of the simplest and best devices is the "Brumbo Pul-

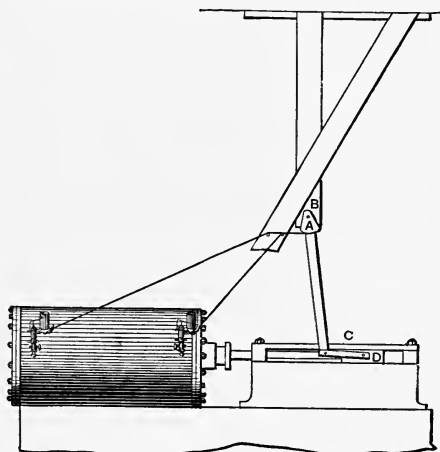


FIG. 29.—THE BRUMBO PULLEY.

ley," Fig. 29. It consists of a sector, *A*, vibrating about an axis, *B*, and actuated by an arm, *C*, and a link, *D*; the latter connected as directly as possible with the cross-head. This may

often be accomplished by attaching its free end to a set-screw on the latter. The sector is usually of wood; but if to be permanently set, is sometimes a light frame of brass. The arm and link are usually of light iron or steel, secured together with nicely fitted pins. The longer the arm in proportion to the stroke of engine-piston, the truer the action; the proportion of two to one should be obtained if it can be done.

The accompanying sketch illustrates a neat device for securing a correct adjustment of the indicator-cord when taking motion from a simple suspended lever. The pin to which the cord is attached is set in a right-angled piece of wood with lines marked upon it parallel to its lower edge and indicating the proper direction of the cord. This is secured on the pendent lever in proper position, when the latter hangs vertically, the engine at mid-stroke, and is then fastened securely by small screws.

A modification of the Brumbo arrangement which the Author has found to work excellently well on high-speed engines is illustrated in the next sketch, as designed originally by Mr. Sweet. When used under the direction of the Author in work done at the Sibley College of Cornell University, it was constructed as follows:

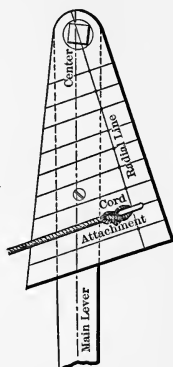


FIG. 30.—CORD ATTACHMENT.

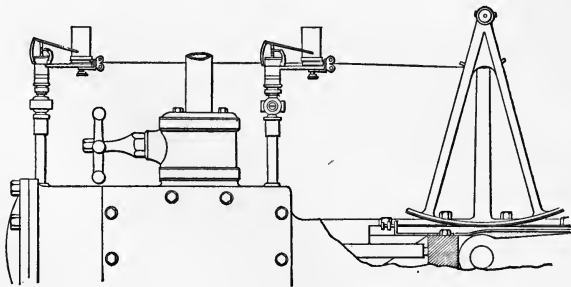


FIG. 31.—INDICATOR ATTACHMENT.

The reducing mechanism used in connecting the indicator-barrel to the cross-head of the engine was fitted with a very firm connecting arrangement, and with an ingenious detaching device. A sector was constructed which was pivoted above the cross-head and hung in the vertical plane above the latter, the engine being horizontal. The arc of the sector carried a pair of steel ribbons, one attached to each end, each carried around the arc and secured, at its opposite end, to the end of a bar fastened on the cross-head, in such manner that, the two ends of the ribbons at the cross-head bar being well secured and tightly drawn up by means of screws placed conveniently for the purpose, all back-lash was prevented, and an absolutely exact synchronism of movement of indicator-line and cross-head was obtained. A smaller sector at the upper part of the larger one was the carrier of the cord, and the combination was thus a perfect means of reproducing the motion of the engine on the smaller scale required in working the paper-barrel of the indicator. The "cord" was piano-wire, a material much less liable to cause difficulty by stretching than any other that was available. Its free part was kept taut by a "spiral" (helical) spring, attached beyond the point of connection with the paper-cylinder.

The cord may either be taken around a groove in the rim of the sector or led from a properly set pin on its side. The latter is the more accurate, pro-

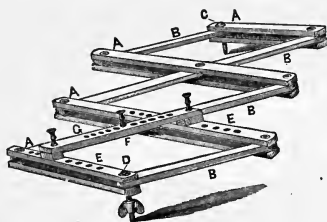


FIG. 32.—THE PANTAGRAPH.

vided the cord at half-stroke is led off at right angles to a radius of the pulley passing through the centre of the pin.

A "pantagraph motion," if well made, nicely adjusted, and properly attached, makes an excellent reducing arrangement. This is seen in Fig. 32.* It consists of a system of levers of wood; those marked *B* are single strips, and

* American Machinist, Dec. 27, 1879.

those marked *A* double strips. The pivot-holes should be bushed. The strip *G* should be arranged so that it may be shifted in the holes *E*, and bring a hitch-pole, *F*, in a line passing through pivots *C*, *D*. The end pivots *C* and *D* should have a projection below, with the end somewhat pointed. The engine cross-head must have a vertical hole in it somewhere, so that pivot *C* can be dropped into it. A stake must be set in the floor near the guides, having a socket for the pivot *D* in its top. Its socket must be level with the cross-head socket, and must be directly opposite the former when the latter is at mid-stroke. The indicator-cord is hooked to the centre peg *F*, and the cord should lead off parallel with the guides.

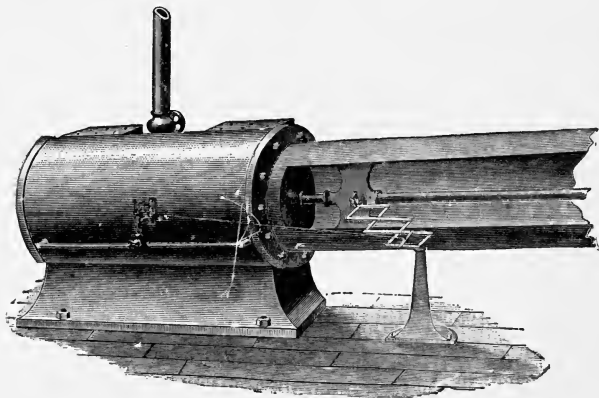


FIG. 33.—THE PANTAGRAPH IN PLACE.

The next illustration shows the apparatus in place, and the indicator attached. Various modifications of this device are in use, all of which embody the same principles.

Fig. 34 shows another form of pantagraph. The working end, *A*, takes motion from the cross-head, and *B* is attached to the floor. The pin, *D*, is fixed in line between the pivot and the working end, and the pulleys, *E*, guide the cords.*

* Barrus on the Indicator.

Avoid the use of long cords. If the motion must be carried a long distance, strips of wood may often be arranged in their place and operated with direct connections. Braided

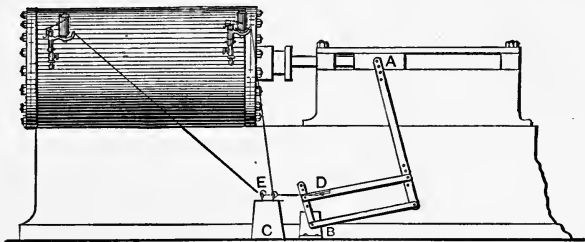


FIG. 34.—THE PANTAGRAPH.

linen cord, a little over one-sixteenth of an inch in diameter, is a suitable material.

The next two engravings exhibit Mr. Thompson's usual methods of attachment for engines of his own design: *A* is the

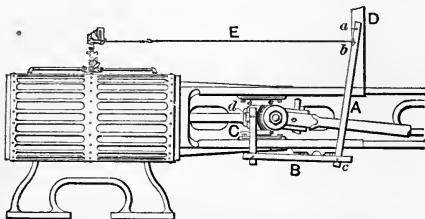


FIG. 35.—INDICATOR "RIG."

lever, *B* the connecting bar, *C* a strip of board attached to the cross-head, *D* a firm support, and *E* the indicator-cord, which is shown horizontal. This horizontal direction allows the pivot *a*, the cord pin *b*, and the pivot *c* to be in line, and when no pipe fittings are used to connect the instrument, and it consequently is shifted from end to end of the cylinder, it is correct for both positions.

Fig. 36 shows a device that will give a movement perfectly

free from distortion. The cord is attached to the end of a short bar, which slides freely in a bearing in the carrying-post. This bar

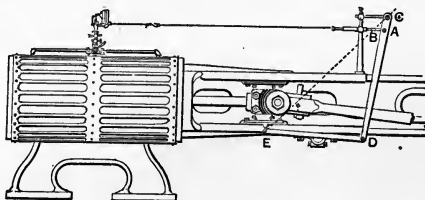


FIG. 36.—INDICATOR ATTACHMENTS.

is connected to the lever CD by a link AB . The lever is connected to the cross-head at E by a bar, DE . The pivots C, B, E are in line at all times; and the distortions of the movement of the lever due to the vibration of DE , will be corrected by the equal vibration of the link AB ; since, $CD : DE :: CA : AB$. This, to be correct, must be proportioned for the engine. The cord should be nearly level.

The cord employed should be as short as possible. If unavoidably long, a fine wire of steel, of iron piano-wire, or of hard-drawn brass should be used. Braided cord is usually supplied by makers of indicators which has been made especially for the purpose and well stretched. A hook on the end of the cord attached to the drum and a loop, Fig. 37, on the adjacent



FIG. 37.—THE LOOP.

end of the cord from the reducing motion afford means of ready connection and disconnection. The loop is adjusted, before hooking on, to just the right length so as to avoid liability of accident by maladjustment when starting. The spring chosen should usually be rated at above one-half the maximum gauge reading; in other words, so that the maximum rise of the pencil may not be above two inches. The minor details of op-

eration are always fully described in the instructions supplied by the maker of the indicator.

Mr. Lyne's method of attachment of the indicator to the locomotive is shown in the accompanying engraving.*

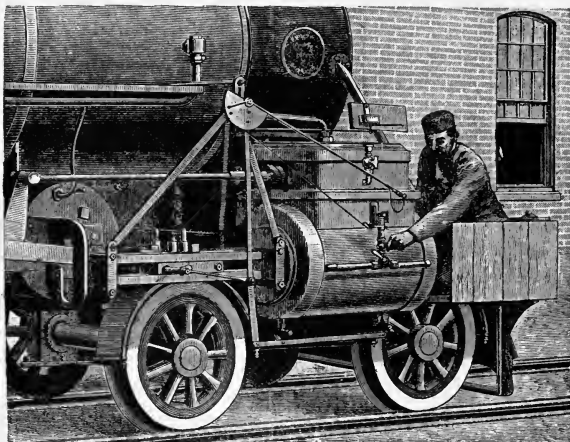


FIG. 38.—“INDICATING” LOCOMOTIVES.

In making connections, a hole is drilled close to the flange at each end and tapped to fit a $\frac{3}{4}$ -inch nipple (wrought-iron pipe). These holes should be so drilled that the piston may not cover the holes at the ends of the stroke. After the holes have been drilled and tapped (which may be done without removing the cylinder casing or back cover), two nipples having $\frac{3}{4}$ -inch malleable-iron elbows upon the ends are screwed in place. These should be tightly screwed up, and set looking towards each other. Two long nipples, having brass collars brazed upon one end, are then cut of a proper length to fill the space between the elbows and cock. Brass should be used for the collars, as solder will not run through the threads if the

* Am. Machinist, Apr. 1, 1882, p. 1.

collars are of iron. The end of one of these nipples is to be cut in the usual way and tightly screwed in place, while upon the other nipple the die is run up $1\frac{1}{2}$ inch for the application of a lock-nut. It is then screwed into the elbow, the three-way cock is put in place, the joints being made with washers of annealed sheet copper. Rubber and all other fibrous material should be avoided in making these joints. The nipple having the lock-nut is screwed out from the elbow until the joints are firmly made upon the cock, then the lock-nut is to be set up against the face of the elbow, a few strands of lamp-wick having previously been wound between the two surfaces to avoid any possibility of leaks. The ends of all the pipes should be carefully rounded inside with a file, and nothing but clean oil should be used in making the joints. If red lead is used, the operator, in taking the diagrams, will have the annoyance of seeing his instrument stick, and will be obliged to remove the piston frequently and pick the lead and dirt out of the instrument.

The three-way cock, shown in the sketch, is of a heavy pattern. The brass centre of the handle has a wooden covering, which protects the hands from being burned. The passages are of quite easy curvature and of large diameter— $\frac{9}{16}$ inch in this case.

The details of this construction are seen in the sketches herewith given.

In applying these devices to a locomotive, place the cross-head at half-stroke, attach the frame to the guides, and set the lever at right angles to the guides and screw up all the nuts, being careful to spread the blocks so that the cross-head will not strike them. Adjust the lever vertically, in the following manner: With a pair of trammels set to the length of the lever, describe an arc, and draw a straight line with the ends touching the arc. Measure from the centre of the line to the arc, and set the centre of the lower pin in the lever half this distance below the centre of the cross-head, then tighten the nut on the upper pin. This will equalize the vibration.

This should all be done beforehand, and the distances put

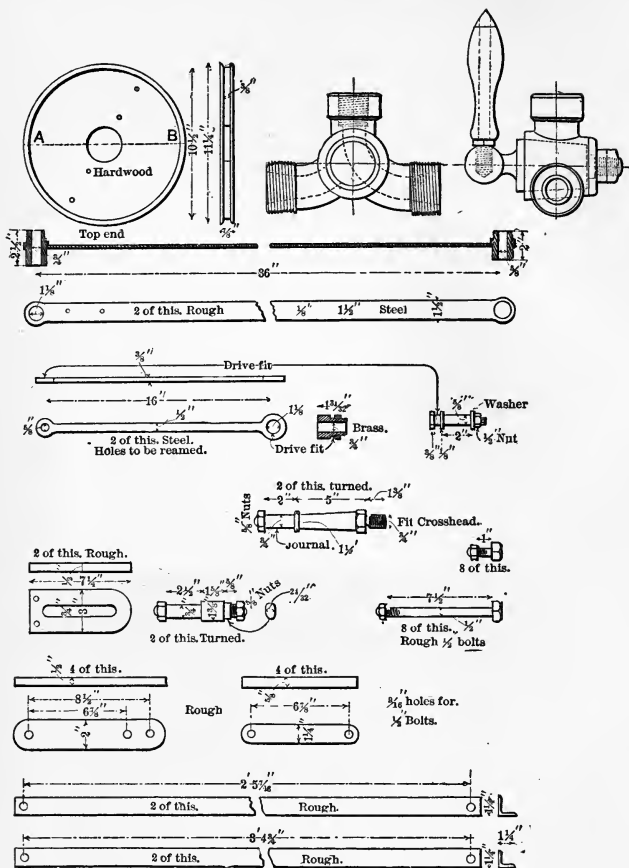


FIG. 39.—DETAILS OF INDICATOR-MOTION.

down in a book for reference. To prevent the cord from getting out of the groove, a wire loop should be used, as shown. This arrangement can be applied and adjusted with-

out moving the engine. The indicator-cord is short, and the lever quite long. The cord is best braided linen line, well stretched and lightly waxed. The box bottom is 9 inches below the beam by a $\frac{3}{4}$ -inch bolt through it and the flag-stand; the back is supported by an iron bracket attached to the stud in the centre of the cylinder-head.

The counter is attached to a board bolted to the front brace, as shown.

In applying an indicator to the steam-chest, drill a hole in the centre horizontally and vertically; then screw in a half-inch nipple and elbow, and set up the indicator in line with the grooved arc. The cord connection will be very short.

Holes should be drilled in every cylinder while the engine is in the shop, and brass plugs, with hexagon heads, screwed into them. The cross-heads should be drilled and tapped, so that upon an hour's notice the indicator may be attached without the necessity of doing any work. In cold weather it is desirable to erect a screen to protect the operator from the wind.

The method of attachment found, on the whole, most convenient, in the work of the Author on vertical marine engines, is that shown in Figs. 40 and 41.* This apparatus was designed for the Author by Mr. Lyne, and used on the steam-yacht "Namouna," while preparing to make some improvements. The engines are of the compound "tandem" type, their cranks at right angles. The high-pressure cylinders are 22 inches in diameter and placed above the low-pressure cylinders, which are 42 inches in diameter with a stroke of 28 inches. The propeller has four blades and a pitch of 18 feet; the boiler-pressure is 80 or 85 pounds per square inch; the engines make from 80 to 85 revolutions per minute. At *A* is the high-pressure cylinder; the low-pressure at *B*; while *C* is the frame, *E* the guides, *D* one of the columns supporting the engine. A wrought-iron arm, *G*, is bolted to the pin on the cross-head *F*. This arm had a rectangular end for the slide *K*. The arm *G* was at right angles to the guides, and, by the aid of a steel steady pin, can be readily removed and replaced, the

* Am. Machinist, Aug. 19, 1882, p. 3.

with but little wear, as the wearing surfaces are all large, and it is always ready for use. It is simple and easily made, and diagrams may be taken in a heavy sea with as great accuracy as in smooth water, as there are no guide-pulleys attached to the woodwork of the vessel.

Two cords for working the indicators are attached to the grooved segment *J*, by passing the end of each through a hole at each side of the groove, as shown at *a a a* in plan, and knotting the ends.

The steam-pipes to the indicators are $\frac{3}{4}$ inch in diameter.

The experience of the Author with this arrangement was thoroughly satisfactory. It is somewhat costly, in comparison with less perfect devices; but its operation is so effective as to fully compensate that disadvantage where, as in this case, it is intended to be a permanent attachment, and kept ready for daily use.

The working drawings of the details of this attachment are presented in the next figure.

A is the lever, *B* the composition sleeve, upon which is fitted the segment *C*. The bearing *D* is turned down in the middle to form an oil-chamber. The bearing *D* is screwed into the collar *K*. The screws for holding this collar together are operated by a screw-driver. The slide *F* is made of composition, having a gib to take up the lost motion. No set-screw is used, as it is safer to insert a liner on top of the gib.

The pin *J* is attached to the slide *F*, and forms a journal for the lever. The thimble *I* is fitted to the pin *J*. *GH* shows the arm bolted to the cross-head.

The following are the details of proportioning several simple indicator-motions devised and described by Mr. Nystrom.*

Fig. 42 represents one of these indicator-motions; though not absolutely correct, the error is insensible. It consists of a horizontal lever, *L*, with its fulcrum at *C*, and the other end attached to the pendulum *P*, the lower end of which is attached to the cross-head. The fulcrum *C* should, as advised by its designer, be placed near the indicator *I*, so as to make the

* Mechanics, June 1883.

lever L of nearly the same length as that of the cord connecting the indicator with the pendulum. The error on the card will then be the difference of the versed-sines of half the angles

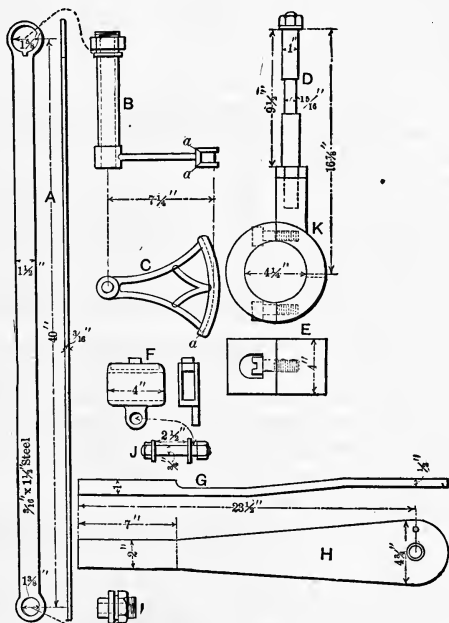


FIG. 41.—DETAILS OF INDICATOR GEAR.

formed by the motions of the lever and cord. All linear dimensions in the following analysis are to be expressed in inches.

L = length of the lever ;

l = length of the cord from the indicator to the vertical position of the pendulum ;

P = length of the pendulum ;

s = half the stroke of the steam-piston ;

ϕ = angle moved by the lever L ;

ϕ^1 = angle moved by the cord;

δ = differential;

v = half the angle of the pendulum motion;

e = vertical motion of the joint of the pendulum and lever.

$$e = P - \sqrt{P^2 - s^2}.$$

Example.—The pendulum $P = 36$, and $s = 12$ inches. Required the motion of the joint.

$$e = 36 - \sqrt{36^2 - 12^2} = 2.0588 \text{ inches.}$$

Half of this motion will then be 1.0294 inches.

Assume the lever $L = 48$ inches, and half the angle ϕ will then be

$$\sin \frac{1}{2}\phi = \frac{1.0294}{48} = 0.021445 = \sin 1^\circ 13' 40''.$$

The versed-sine for this angle is $0.00023 \times 48 = 0.011$ of an inch.

$h = 35.5$ inches vertical mean height of the cord above the direction of the stroke s ; then the versed-sine will be reduced

to $\frac{h}{P}$, or $\frac{0.011 \times 31.5}{36} = 0.009625$ of an inch.

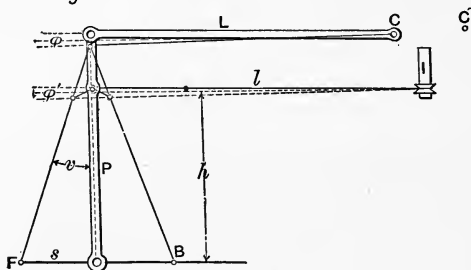


FIG. 42.—INDICATOR-MOTION.

The vertical motion of the cord at the pendulum will be $\frac{2.0588 \times 31.5}{36} = 1.80145$ inches, of which one-half = 0.900725.

Half the angle of the cord will then be

$$\sin \frac{1}{2}\phi' = \frac{0.900725}{48} = 0.00187765 = \sin 1^\circ 4' 30''.$$

$$l = 54 \text{ inches.}$$

The versed-sine for this angle is $0.00017 \times 54 = 0.00918$ of

an inch. The error on the card caused by this indicator-motion will then be $0.009625 - 0.00918 = 0.000445$ of an inch. This error is too small for detection on the indicator-card, but it can be removed entirely by placing the fulcrum C at C' on the other side of the indicator. Fig. 42 can therefore be considered a reliable indicator-motion.

The letters F and B represent the position of the pendulum when the piston is at the front or back of the cylinder.

Fig. 43 represents another of these indicator-motions for the locomotive. It is similar to that of Fig. 42, except that the

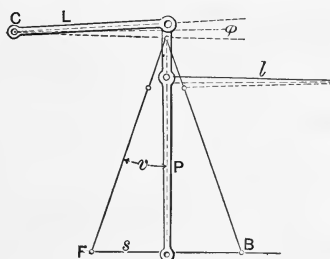


FIG. 43.—INDICATOR-MOTION.

lever L is placed at the other side of the pendulum, as a matter of convenience. There were two cylinders to be indicated simultaneously, for which purpose horns, ab , Fig. 44, were fixed on the pendulum from which cords were led. The pendulum was of steel, 2 inches wide by $\frac{3}{8}$ inch thick, and the pivot-holes were $\frac{1}{2}$ inch. The ends of the pendulum were made 3 inches in diameter for the purpose of making it firm against uneven action on the horns, which were $\frac{3}{4}$ inch in diameter at the pendulum and tapering to $\frac{1}{4}$ inch at the eyes. This made a very rigid system, which worked well. The fulcrum C for the lever L was fixed on the foot-board of the locomotive. The pendulum was 36 inches, lever 24 inches; and stroke of steam-piston 24 inches, stroke of card 3 inches. The calculation for error is the same as that for Fig. 42, except that the versed-sine of the cord must be added to that of the lever L . The error so obtained was 0.018, which, divided at each end of the card, makes it 0.009. This error exists at $\frac{1}{8}$ from each end of the card, positive at one end and nega-

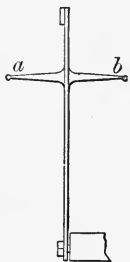


FIG. 44.—HORNS ON INDICATOR-MOTION.

tive on the other; but when cards are taken from both ends of the cylinder the errors compensate.

The angle v of the pendulum at the ends of the stroke is

$$\sin v = \frac{s}{P} = \frac{12}{36} = 0.3333 = \sin 19^\circ 28'.$$

The indicator-motion, as represented by Figs. 45, 46, and 47, is a defective indicator-motion.

Fig. 45 exhibits the characteristics of link indicator-motions. The cross-head moves in the direction of the dotted line FB ; the link L is made very short in proportion to the length of the pendulum P , for the purpose of better illustrating the motion. The different positions of the link and pendulum are numbered 1, 2, 3, etc., of which in the first position the link and the

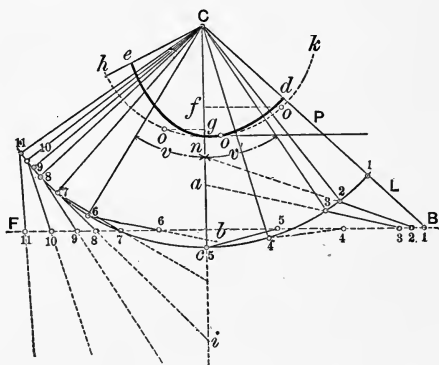


FIG. 45.—LINK INDICATOR-MOTION.

pendulum are in a straight line; the cross-head is stationary while the pendulum moves. The line Cc is the vertical position of the pendulum. The point on the vertical Cc , where the direction of the link crosses, shows the motions of the cross-head and of the pendulum; the motion of cross-head is to pendulum as Ca is to Cc . In the fifth position the link

crosses the vertical at c ; the motions of the pendulum and cross-head are alike. In the eighth position the line of the link crosses the vertical at i ; the motion of cross-head is to pendulum as Ci is to Cc .

On the right side of the vertical the pendulum moves faster than the cross-head, and on the left side the cross-head moves the faster. When the link is less than half the length of the pendulum, the latter should move over a much smaller angle on the link side than on the other side of the vertical. The motion of the pendulum transmitted to the indicator by a circle-sector, h, n, k , will make the cord move too fast at the ends of the stroke. A sector should not be circular, but of the form e, n, d , for the proportion of link and pendulum shown in Fig. 45; then,

$$C, e : C, n = C, a : C, c,$$

$$C, d : C, n = C, b : C, c,$$

$$C, e : C, d = C, a : C, b.$$

The indicator-cord fixed on the pendulum, without a sector, will move too fast on the link side and too slow on the other side at the end of stroke. The cord should be fixed at o , on the link side of the vertical, and so that

$$C, f : C, n = C, a : C, c,$$

$$C, g : C, n = C, b : C, c,$$

$$C, f : C, g = C, a : C, b.$$

When the cord is attached to the sector h, n, k , or to the pendulum at n , the pendulum should make a smaller angle on the link side and a greater angle on the other side, but it will not make a true card in any case.

Fig. 46 represents a motion very much used. The direction of motion from the cross-head bisects the versed-sine of the angle made by the pendulum, and the angles v are alike on both sides of the vertical.

Here,

$$P = 18 \text{ inches;}$$

$$L = 8 \text{ inches;}$$

$$e = 4 \text{ inches;}$$

$$h = 17\frac{1}{8} \text{ inches;}$$

$$v = 26^\circ 23';$$

$$S = 16 \text{ inches;}$$

$$Ca = 15.05 \text{ inches.}$$

From these data we find that the indicator will move 14 per cent. faster at the back than at the front end of stroke.

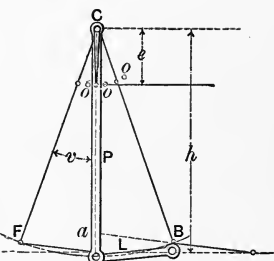


FIG. 46.—INDICATOR-MOTION.

Lengthening the link, correct the defect in part, and the motion would make a tolerably correct card.

Fig. 47 represents a motion in which

$$P = 48\frac{3}{8} \text{ inches, length of pendulum;}$$

$$L = 34\frac{1}{8} \text{ inches, length of link;}$$

$$h = 38 \text{ inches, height of fulcrum;}$$

$$v = 30^\circ, \text{ half angle of motion;}$$

$$S = 48 \text{ inches, stroke of piston;}$$

$$Ca = 44\frac{1}{4} \text{ inches;}$$

$$Cb = 39 \text{ inches.}$$

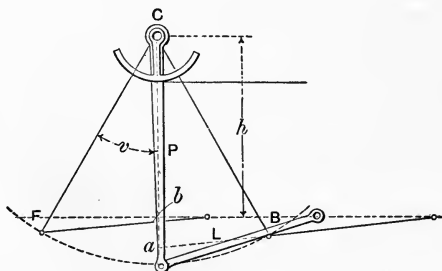


FIG. 47.—INDICATOR-MOTION.

The indicator would move nearly 14 per cent. faster at the front than at the back end of stroke. The link should be

made 29 inches long, to make the angle of motion greater on the link than on the other side the vertical. The motion would then produce a better card.

Slot motions for indicators have been tried, and abandoned.

Fig. 48 represents a motion in which the cord is moved by a circle-sector.

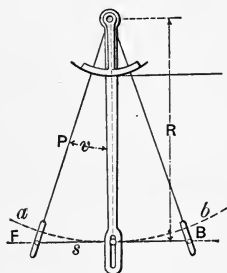


FIG. 48.—SLOT INDICATOR-MOTION.

The slot-pin is fixed in the cross-head of the engine, and the problem is to find the relative motion of pin to slot on the arc ab . The height R of the fulcrum is constant.

P = distance from the fulcrum to any position of the slot-pin ;

S = half the stroke of the steam-piston ;

a = half the length of the arc ab .

$$\tan v = \frac{R \sin v}{\cos v}.$$

Differentiate this formula, and we have

$$\delta \tan = \frac{\cos \delta \sin - \sin \delta \cos}{\cos^2},$$

$$\delta \tan = \frac{(\sin^2 + \cos^2) \delta a}{\cos^2},$$

$$\delta \tan = \frac{\delta a}{\cos^2},$$

and

$$\delta a = \cos^2 \delta \tan.$$

The differential of the tangent is the differential motion of the pin, which we may take as unit,

$$\delta a = \cos^2.$$

The motion of pin is to arc motion as $1 : \cos^2$.

If radius $R = 36$ inches, and $s = 15$ inches, v will be

$$\tan v = \frac{s}{R} = \frac{15}{36} = 0.41666 = \tan 32^\circ 37'.$$

The cosine for this angle is 0.9231.

The square $0.9231^2 = 0.8521$, and when the motion of the pin is 1, near the end of the stroke, that in the arc at a will be 0.85, 15 per cent. too slow.

The error may be corrected by converting the sector into a triangle, as shown by the dotted line.

Fig. 49 represents a motion in which the cord is fixed without a sector. In this case $\delta \sin$ must be inserted for δa ; then

$$\delta \sin = \cos \delta a.$$

$$\delta a = \frac{\delta \sin}{\cos}.$$

Insert this for δa , and we have

$$\frac{\delta \sin}{\cos} = \cos^2, \text{ or } \delta \sin = \cos^3.$$

The motion of pin is to cord as $1 : \cos^3$. Let $R = 36$, $s = 15$, $v = 22^\circ 37'$ and $\cos v = 0.9231$; $0.9231^3 = 0.7865$. When the motion of the pin is 1, near the end of the stroke, that of the cord will be then 21 per cent too slow. Slot motions distort the card at the ends of stroke, where greatest accuracy is required for exhibiting the method of distribution of steam.

56. Precautions essential to the successful employment of the indicator have been already detailed at some length; briefly summarized, they are: (1) Make sure of accuracy of construction of the instrument in its dimensions and fitting; (2) Secure exactness in scale of the spring employed, not when cold, but when hot and in use; (3) Demand the utmost lightness and stiffness of moving parts; (4) See the spring in the paper-drum correctly adjusted to the speed of the engine; (5) See that the instrument is well lubricated, and with the best of light oils, and that it works freely and without friction; (6) Make the steam-connections short, straight, and large; (7) Use a short cord, and substitute wire where any considerable length is necessary; (8) See that the reducing motion is perfectly accurate, free

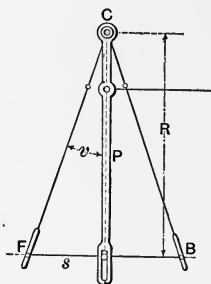


FIG. 49.—SLOT MOTION.

from lost motion, and both strong and light ; (9) In taking the diagrams, see the steam-cock opened full, the indicator well heated up, the steam condensed in the connections completely blown out, and the touch of the pencil on the paper as light as is consistent with making perfectly legible diagrams.

Every maker gives detailed instructions for care of the instrument and for its dissection and assemblage. The principal points are the following, details varying with the style of the indicator :

Before using any indicator, take it apart, clean, and oil it. Try each part separately. See if it works smoothly ; if so, put it together without the spring. Lift the pencil lever, and let it fall ; if perfectly free, put in the spring, and connect. Give it steam, but do not attempt to take a card until it blows dry steam through the relief openings. If the oil from the engine gums the indicator, take it off and clean it.

Never use red or white lead in connecting, as it is liable to get into the instrument. Attach the indicator to the cock by coupling the differential threads of the indicator shank and cock. The lighter the spring used, the higher will be the diagram produced, and the more accurate the measurements obtained ; in selecting a spring, choose one to give diagram about two inches high.

After the desired number of diagrams have been taken, remove the piston, spring, etc., from the indicator, while it is still upon the cylinder ; allow the steam to blow for a moment through the indicator cylinder ; then examine piston, spring, and all movable parts, which must be thoroughly wiped, oiled, and cleaned. Particular attention should be paid to the springs, as their accuracy will be impaired if they are allowed to rust ; and great care should be exercised that no gritty substance be introduced, to cut the cylinder or the piston. The springs should not be left in the indicator. The pencils can be best sharpened with a fine file.

Each blank indicator-card usually has printed on its back a set of data to be filled out, such, for example, as the following. The number and character of these items differ in the practice

of different engineers; but the more important are never omitted.

| | |
|--------------------------------------------|----------------------------------|
| Date | |
| Diagram from | Engine.....End.....No..... |
| Diameter of Cylinder and Area | Built by H. P. Factor.... |
| " " Rod and Area..... | Initial Pressure.....I. H. P.... |
| Length of Stroke..... | Brake H. P..... |
| Clearance and..... | Barometer reads..... |
| Revolutions per Minute | Thermometer..... |
| Piston Speed | |
| Pressure of Steam, in lbs., in Boiler..... | |
| Point of Cut-off | |
| Position of Throttle-valve | Observer..... |
| Vacuum per Gauge, in inches | Remarks. |
| Temperature of Hot-well | |
| " " Injection..... | |
| Scale of Spring | M. E. P..... |
| Inside Diameter of Feed-pipe..... | |
| " " " Exhaust-pipe..... | |
| Valves | |

The Author has used the next form many years. It is amply complete for most cases; indeed is rarely entirely filled out.

| | | |
|-------------------------|-----------------------------------------|--------------------------------|
| Time.....Date..... | R. H. THURSTON, CONSULTING ENGINEER. | Builder of Eng. { |
| Diam. of Cylinder..... | | Kind of Valve Motion..... |
| Length of Stroke | Owner of Engine: | " " Steam-valves |
| Revolution per min..... | | " " Exhaust-valves..... |
| Speed of Piston | | " " Condenser |
| Diam. Piston-rod..... | Kind of Work Driven by Engine: | " " Heater..... |
| Area Steam-port. | | " " Boiler..... |
| " Exhaust-port | | " " Fuel |
| Piston Clearance..... | | Temperature of Feed-water..... |
| Port " | | " " Hot-well..... |
| Boiler Pressure | Remarks: | Water per hour. |
| Initial " | | Coal " " |
| M. E. Pressure | Barometer | |

CHAPTER V.

INDICATOR-DIAGRAMS INTERPRETED.

57. Indicator-diagrams taken under proper conditions and with good instruments are diagrams of energy on which the ordinates measure the varying pressures in the cylinder, corresponding to the positions of the piston as measured off by the simultaneous abscissas of the diagram; while the area represents the work done by the steam on the piston of the engine. The forms and relations of the several lines of which the diagram is composed reveal the method of action of the valves and the effectiveness of the pipes and passages as conduits for the entering and the exhausted steam. The correct interpretation of the diagram thus becomes an exceedingly important matter.

58. The Typical Diagram and its Nomenclature, assuming the indicator applied to the steam-engine, are as below.

The curves described on the indicator-cards of engines present many differences as to the mode in which pressure and volume vary, and their figures cannot be expressed by any mathematical formulæ; since it is impossible to separate those irregularities which arise from fluctuations in the pressure of the steam from those which arise from the friction and inertia of the moving parts of the indicator, and also because the law of such changes as actually take place in the cylinder of the engine is not precisely known.

An approximate form of diagram is therefore taken in theoretical treatment, which diagram is approached more and more closely as the machine is improved. Fig. 50 is such a diagram. *AB* represents the volume of the mass of steam

when admitted into the cylinder. The *first assumption* is that the pressure of the steam remains constant during admission, so that AB is a line parallel to OX , and the pressure is represented by $OA = GB$. The *second assumption* consists in assigning to the curve BC one or other of two definite forms:

(I) When the cylinder has no steam-jacket, the steam is assumed to expand without receiving or giving out heat; so that BC is an *adiabatic curve*.

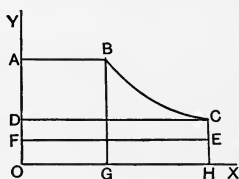


FIG. 50.—IDEAL DIAGRAM.

(II) When there is a steam-jacket, it is assumed that the heat communicated by means of that jacket is just sufficient to prevent any appreciable part of the steam from becoming liquid; so that BC is a curve of pressures and volumes of saturated steam.

The real diagram of the ordinary engine is of somewhat different form. The next figure illustrates these differences. The accepted nomenclature is as follows:

The admission line AB is produced by steam on admission. Its normal direction is vertical, or nearly so, as it is traced while the crank is passing its dead-centres. Leaning outward indicates lead. With no lead it would lean inwards, as sometimes with condensing engines.

The steam line BC is traced after the piston has commenced its stroke. Its proper direction is horizontal at a pressure nearly equal to that in the boiler; but this can only be approximated with such openings that the maximum velocity of flow will not exceed about 100 feet per second. But with throttling-engine diagrams the steam line inclines downwards.

The point of cut-off C is the point where the entering steam is cut off. It is usually anticipated by a fall of pressure, which is less as the valve closes more promptly. With some engines, having multiported gridiron valves with detachable valve-gear, this fall of pressure is not appreciable, and the point of cut-off is well defined.

When the instrument has been in good working order, the cut-off may be located at the point of contrary flexure.

The expansion line CD begins at cut-off and terminates at exhaust.

The point of exhaust D is where exhaust begins; the expansion curve there ends and the pressure begins to fall rapidly.

The exhaust line DE is traced while the steam is escaping. When it occupies a considerable part of the return stroke, or nearly all, it indicates a cramped exhaust opening.

The back-pressure line EF represents the pressure in the

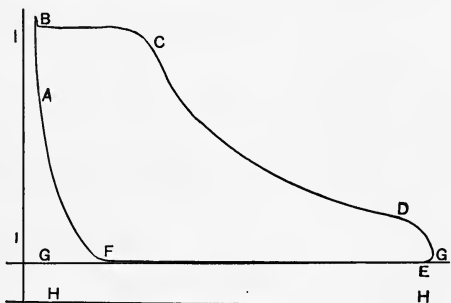


FIG. 51.—NOMENCLATURE OF CARDS.

cylinder during the return stroke. With non-condensing engines the position of this line is somewhat above atmospheric pressure. With condensing engines it indicates a pressure somewhat in excess of that in the condenser.

The point of exhaust closure F is anticipated by a rise of pressure; the eye may locate it very exactly.

The compression curve FA exhibits the method and extent of variation of pressure after the exhaust-valve closes.

The atmospheric line GG locates the position of equilibrium of the piston of the indicator before steam is introduced.

The vacuum line HH is drawn parallel with the atmospheric line at such a distance below it as will measure the pressure of the atmosphere. It is generally placed 14.7 lbs. below the atmospheric line. When a barometer can be con-

sulted, its reading in inches, divided by 2, will give approximately the atmospheric pressure.

The next figure illustrates the form of the diagram obtained from an explosive gas-engine of the Otto type. The three lines, *ABC*, are the result of three successive explosions with varying rates of combustion, *A* indicating rapid, and *C* show-

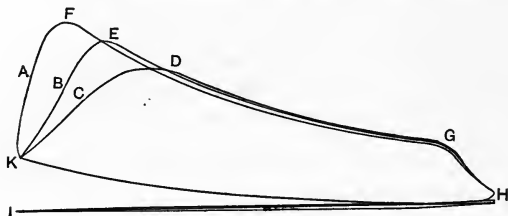


FIG. 52.—GAS-ENGINE DIAGRAMS.

ing slow, combustion; neither representing a true explosion, which would have given an initial line above *A*, and vertical. Here the mixture of gases with air enters on the induction-

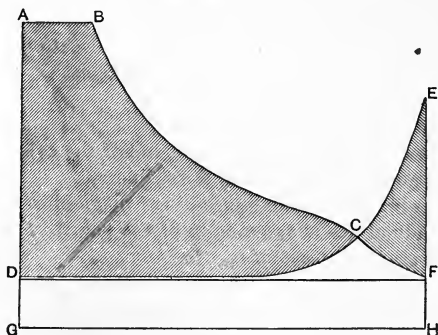


FIG. 53.—EFFORTS ON THE PISTON.

stroke *IH*; compression occurs on the return of the piston, *HK*; explosion follows and a second out-stroke, *KFG*; exhaust takes place at *G*; and expulsion of the charge of non-consumed gases takes place on the second return-stroke, *HI*.

The indicator-diagram, although generally assumed to represent the variation of the effort on the piston of the engine at each half-revolution, really exhibits only one part of that action at any given instant. The line $ABCF$, Fig. 53, exhibits the effort of the steam during the forward stroke; but that effort is partly equilibrated by the back-pressure and the compression on the opposite side. If these are represented by the line DCE , it is evident that the real variations of net effort are exhibited by the space $ABCD$ and by CEF ; the former being positive, the latter negative. It is thus necessary to combine parts of two opposite simultaneous diagrams to ascertain the real pressures transmitted from the piston.

59. The Causes of Modified Forms of Diagram are usually simple and easily traced. The actual form of the diagram differs from the ideal form, as just described, in consequence of the occurrence of a number of conditions which are usually more or less objectionable. These conditions are classed thus:

Causes which affect the power of the engine, as well as the figure of the diagram:

- (1) Wire-drawing in taking steam and at cut-off.
- (2) Clearance in the cylinder and passages.
- (3) Compression, or cushioning.
- (4) Pre-release.
- (5) Conduction of heat by the metal of the cylinder.
- (6) Liquid water present in the cylinder.

Causes which affect the figure of the diagram only:

- (7) Undulations in the motion of the pencil.
- (8) Friction of the indicator.
- (9) Position of the indicator.

In the accompanying sketch, in which the ideal and a modified form are compared, it is easy to trace some of the causes of difference.

At A the pressure of steam is usually a maximum. Should the induction occur at the right time and in the right way, the cylinder will be full of steam at the instant of forward movement of the piston; should the valve open late, A will be found

the real indicator-card. This action is called "wire-drawing" the steam. Where the corner at cut-off is obscured in this way, the real point of cut-off may be approximately determined by carrying out the lines *AH* and *PG* to their intersection at *B*, which is taken at the point required. This has been called the point of virtual, or effective, cut-off.*

Wavy lines indicate a defect in the indicator, or its inapplicability at such speeds of engine. They do not always give rise, however, to inaccurate computations. Broken and irregular lines indicate the presence of grit in the instrument.

The accompanying fac-similes of cards taken from a "high-speed" engine well illustrate the method of variation of the diagram, with loads varying from overload to simple friction of engine.

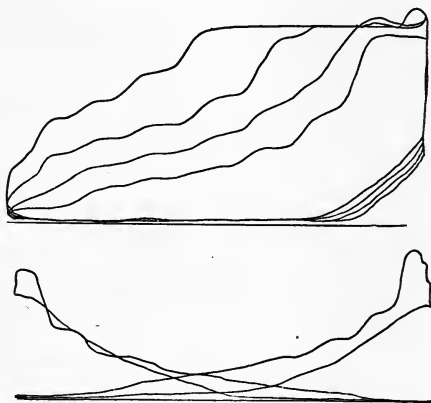


FIG. 55.—VARIATIONS OF LOAD.

The data relating to this case are as follow:

Diameter of cylinder . . . = 8"

Stroke of piston . . . = 10"

Scale, 60 lbs. to 1 inch.

Revolutions . . . = 340 per minute.

* Rankine; Steam-engine, p. 418.

| | | |
|-------------------------------|------------|------------------|
| Weight of reciprocating parts | = | 152 lbs. |
| Connecting-rod length . . . | = | 6 cranks |
| Maximum valve travel . . . | = | $2\frac{5}{8}$ " |
| " lead | = | $\frac{1}{8}$ " |
| " port-opening . . . | = | $\frac{1}{2}$ " |
| Clearance, each end, . . . | = | 11% |
| Maximum, crank end, { | M. E. P. = | 61.80 lbs. |
| | H. P. = | 53.76 |
| " front " { | M. E. P. = | 69 lbs. |
| | H. P. = | 60.03 |
| " mean, { | M. E. P. = | 65.40 lbs. |
| | H. P. = | 56.89 |
| Average initial pressures . . | = | 80 lbs. |

60. The Interpretation of Diagrams is usually easily effected, and by means of this "engineer's stethoscope" it becomes possible to ascertain the nature and cause of almost every defect in the distribution of pressures and volumes of the working fluid, as in the adjustment of the valve-motion, and the size or proportions of steam-passages, or of the connecting pipes. The power exerted by the steam is easily measurable. These several points may be summarized thus:

- (1) Gross power exerted by the steam.
- (2) Net power of the steam, and equivalent net power of the engine.
- (3) Resistance of unloaded engine.
- (4) Net power of the engine.
- (5) Details of various wastes of power, as by wire-drawing, back-pressure, etc.
- (6) Valve-adjustments.
- (7) Effectiveness of valve-gearing.
- (8) Adequacy of sizes of port.
- (9) Quantity of steam present at any point in the stroke.
- (10) Feed-water demanded, exclusive of that wasted by cylinder condensation.
- (11) With a boiler-trial, the actual expenditure of steam, fuel, and money, for a given amount of power; and wastes by leakage and condensation.

Of these, the principal are only determined by careful computation, employing as data the quantities graphically measured on the indicator-diagram; others are at once seen by the practised eye, demanding only an inspection of the figures shown on the card. An engine well adapted to its purpose, a perfect engine in the engineer's sense, will usually exhibit an early induction; wide port-opening; an admission-line closely approaching boiler-pressure, and nearly or quite horizontal;

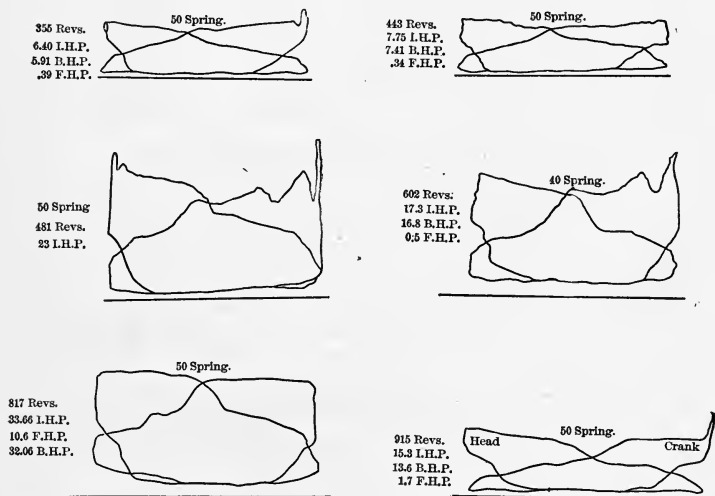


FIG. 56.—EFFECT OF SPEED.

a sharp cut-off; an expansion-line closely approaching the common, or equilateral, hyperbola in form; a somewhat early and a prompt release or exhaust; a low and uniform back-pressure; and a compression carried up well toward initial pressure. These effects are obtained by giving some steam and exhaust lead—greater as speeds and pressures are higher—having good area of ports, securing quick action of the expansion-valve, and a well-adjusted closure of the exhaust-valve. Any depar-

ture from these conditions is ordinarily to be taken as evidence of defective construction or adjustment.

The reduced copies on the opposite page show how the ideal diagram is departed from in the operation of engines, especially at high speeds (Fig. 56).*

All these cards were taken with the same indicators and from the same engine. All exhibit similar departures from the ideal form of diagram, and all illustrate well the two kinds of effect already described—those due to practical conditions of construction and operation of the engine, and those produced by the inertia and friction of the indicator.

As an illustration of the interpretation of the diagram, we may take the following example (Fig. 57):†

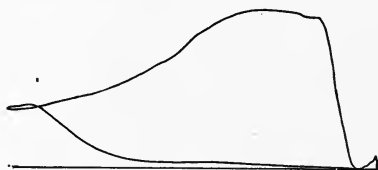


FIG. 57.—NEGATIVE LEAD.

In this case the eccentric sheave had been given, instead of the usual angular advance, a reversed position 43 degrees behind its proper location on the shaft.

The admission commences only when the piston has trav-



FIG. 58.—A GOOD DIAGRAM.

elled one-sixteenth of the stroke. The release is late by an

* Variable Load, etc., R. H. Thurston ; Trans. Am. Soc. M. E., 1888.

† Barrus on the Indicator, p. 19.

average similar amount. The pressure before cut-off is low, the back-pressure high, and there is no compression.

Waste of power is here evidently produced; the expansion is too early terminated and the exhaust is deferred, wasting steam in even higher degree than power of engine. The succeeding figure represents as close an approach to the ideal form as is often seen, and probably has too high a ratio of expansion to give best results.

The accompanying diagram, taken by Mr. King * from the condensing engines of the "Powhatan," and the dotted variations from the actual line, exhibit again the various principal deductions to be made.

Fig. 59 is what would be termed a good diagram.

Steam..... 10
 Vacuum.... 27
 Hot-well..... 106 Fahr.
 Revolutions.... 9.5
 Throttle..... 8.

"Powhatan" stb. cylinder, bottom.
 Nov. 7, 1855, 10 A.M.
 One engine and one wheel in
 operation.
 Smooth sea.

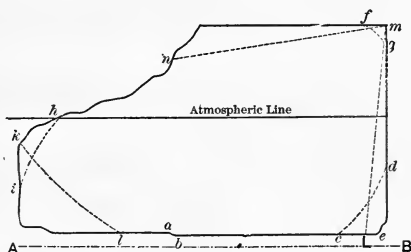


FIG. 59.—TYPICAL DIAGRAM.

It appears, however, that the piston of the indicator worked tightly, which occasioned it to stick in places, as is evidenced by the steps in the expansion-line, and also at *ab* in the vacuum-line.

Should Fig. 59, instead of as shown, have the lower right-hand corner cut off as at *cd*, the exhaust-valve closed too soon,—at *c* instead of *e*,—occasioning cushioning.

* Practical Notes, p. 46.

Had the upper right-hand corner been as shown by the line *fg*, the steam-valve must have opened too late. Had the exhaust corner been cut-off, as shown by *hi*, the exhaust-valve would have opened too soon; but had it been at *kl*, it would have opened too late, and would move too slowly, preventing free escape of steam; or the exhaust-passages would have been too small, which would produce a similar effect. Had the steam-line fallen as at *mn*, it would have shown that the throttle was partially closed, or the steam-passages too small. Should there be excessive lead to the steam-valve, the line *dm* will have the top inclined to the right as *L* from *to m*. Late opening would produce an inclination in the opposite direction.

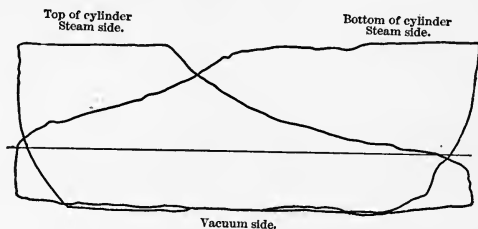


FIG. 60.—UNSYMMETRICAL VALVE-SETTING.

This figure is a double diagram taken from one of the paddle engines of the "Great Eastern," when on her trial-trip in the British Channel.

It will be observed that the valves were unevenly set. The diagram from the top of the cylinder shows that the pressure on the piston was 20 pounds, cut-off at one-third the length of the stroke, and expanded down to atmospheric pressure at the termination. The diagram from the bottom of the cylinder shows that the steam was at 22 pounds, cut-off at half-stroke, and expanding to 4 pounds above the atmosphere at the termination; in both cases the vacuum being 12 pounds or about 24 inches. The number of strokes was $11\frac{3}{4}$, and the speed of the piston 331 feet per minute. The exhaust closed when the piston had travelled five-sixths of the length of the cylinder,

checking the progress of the piston when about two feet from the end of the cylinder.

Whenever the adjustment of valves is proposed, guided by the indicator, it should be carefully noted whether parts affected by such adjustment are liable to injury by the change. Slide-valves, for example, which have been long at work, sometimes wear their seats to a shoulder when they are not terminated by a depression which is overrun by the valve. If this shoulder is not first removed, the change may cause leakage, or even accident.

61. Compound-engine Diagrams, as produced by the steam-engine indicator, often differ even more than with the simple engine, from that ideal "card" which would be given were the expansion precisely as intended, and the engine free from defect, as in clearance-spaces and port-resistances. The best compound engines show considerable loss, as has been seen, in these ways, and also in that drop of pressure between high- and low-pressure cylinders which often constitutes a very sensible source of waste of heat, steam, and fuel. Where the Wolff system is adopted, however, if the load be constant and the machine well proportioned to its work, and if the "dead-spaces" can be made small, the approximation of the actual to the ideal card may be very close, as is illustrated by the accompanying pair of diagrams from a pumping-engine of this character.

The action of the steam and its variations of pressure are here seen throughout the cycle to be precisely similar to that in a simple engine. Large steam-ports and a good expansion-gear bring the steam-line close up to that of boiler-pressure; a well-jacketed cylinder allows the expansion-line to follow closely that laid down for the ideal engine; short and free ports between the two cylinders give an exhaust from the high-pressure and a supply to the low-pressure cylinder which are nearly coincident; and the two cards would, if reduced to a single diagram, exhibit a very close approximation to that which would have been constructed as the ideal diagram of this class of engine. Such satisfactory results are rare; and in most

cases the differences between the actual and the ideal case are very marked, and are serious in their effect upon efficiency.

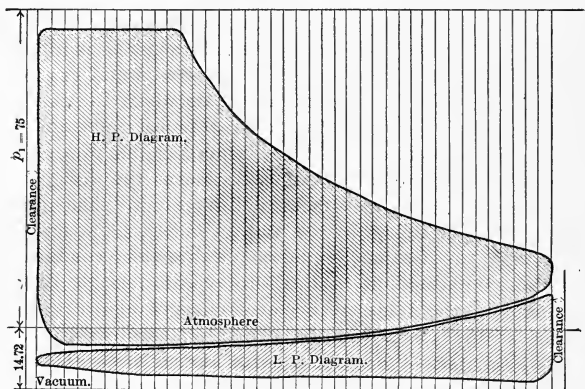


FIG. 61.—ACTION OF STEAM; WOLFF ENGINE.

The next diagrams represent the pair of cards taken from a well-known single-acting compound engine of small size. The

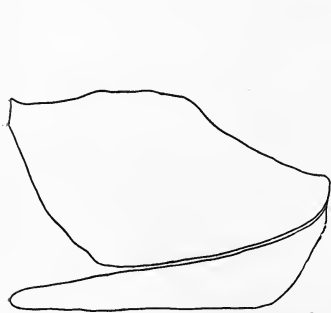


FIG. 62.

COMPOUND-ENGINE CARDS.

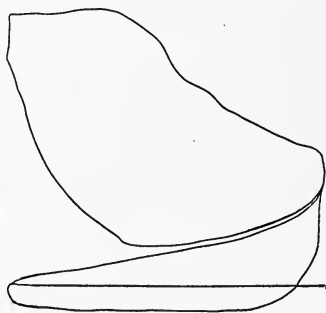


FIG. 63.

action is precisely as before, except that heavy compression is introduced to fill the comparatively large clearance-spaces.

action is represented by DA and EF ; the ordinates of DA representing the backward pressures in the small cylinder, and EF the forward pressures in the large cylinder.

During return stroke of the larger cylinder the steam is expelled, exerting a back-pressure along FA ; while steam is admitted again into the small cylinder, and expanded during a new stroke of that cylinder.

Thus are obtained the diagrams $BCDAB$ for the small cylinder and $EFAE$ for the large, and the sum of their areas represents the energy exerted by the quantity of steam expended.

When the diagrams are to be used for the purpose of studying the relations between heat expended and work performed, it is best to combine them into one diagram, thus:

Draw a line KGH parallel to POQ , intersecting both diagrams, and lay off upon it $\overline{HL} = \overline{KG}$; and $\overline{GL} = \overline{GH} + \overline{KG}$ represents the total volume in both of the steam-cylinders, when its pressure is \overline{OG} ; while L is a point which would have been reached had the action taken place in the large cylinder alone.

By drawing a number of lines, as KL , any number of points may be found to complete the *combined diagram* $BCDLMAB$, whose length $\overline{OQ} = \overline{OP}$ represents the volume of the large cylinder; and this diagram may be discussed as if it represented the action of the steam in the large cylinder only.

Thus, as observed by Rankine,* *the energy exerted by a given portion of a fluid during a given series of changes of pressure and volume depends on that series of changes, and not on the number and arrangement of the cylinders in which those changes are undergone.*

When the diagrams taken from the two cylinders of the simplest form of compound engine are placed together, having the same length, they form a whole such as is illustrated in Figs. 61, 62, and 63. Since, however, the diagram should be constructed so as to make the horizontal scale one of volumes, the pair cannot be compared with the card taken from a simple engine, and they must be reduced to a common scale of vol-

* Steam-engine, § 261, p. 336.

umes by either reconstructing the smaller on the volume scale of the larger, or *vice versa*. The latter is the usual course, as is illustrated in the succeeding examples of compound-engine diagrams. In thus combining these diagrams, it must be remembered that the work represented is that of the engine for one stroke or is that of a single charge of steam, just as in the single cylinder. The base-line measures the stroke, or the volume swept through by the piston in one forward movement. To make a combined diagram, each abscissa of the low-pressure diagram must be increased in such a proportion as to cause it to become proportional to the total volume assumed by the steam at the instant of the production of that line. The ratio of the enlargement is that of the effective capacities of the two cylinders. If the diagrams have both been taken with the same indicator-spring, the two diagrams may now be adjusted to make a single one which exhibits, at all periods of its cycle, the actual relations of pressure and volume of the steam. Were the engine perfect in its proportions and its operation, the two figures would produce a combined diagram precisely like that which might be obtained from a single engine working the same weight and pressure of steam with the same expansion-ratio.

At any point in the simultaneous motion of the two pistons the volume of the steam will evidently be the sum of the volumes, v_x traversed by the large, and V_x still to be moved through by the small, piston; and if the part of the volume traversed in each cylinder is x , this total volume, as to x , is

$$v - x + \frac{V}{v} x = v_x + V_x;$$

and the pressure at this point is measured by the ratio of the volumes; so that

$$p_x = \frac{p_1 v \div r_1}{v_x + V_x} = \frac{p_1 v}{r_1(v_x + V_x)} = \frac{p_1 v}{r_1 v + (r - r_1)x}.$$

This value is somewhat modified by the presence of the intermediate passages between the cylinders, a drop occurring in the pressure at the instant of opening the exhaust from the small cylinder; but this drop is less as those passages are larger; and if forming an intermediate reservoir, as is sometimes the case where "reheating" between the cylinders is practised, this loss and the corresponding reduction in the mean pressure obtained, in work done, and in the actual total ratio of expansion, is sometimes quite unimportant compared with the gain by that process. A common value for the reduction of total expansion is not far from twenty per cent., rising to one-third with small reservoirs and falling to a lower figure with larger spaces. The loss of work may usually be neglected.

The receiver type of engine with equidistant cranks and intermediate reservoirs is less seriously affected by intermediate spaces. The reduction of pressure and the loss of total expansion is but about ten per cent., where the receiver-space is equal to the volume of the smaller cylinder, and falls to less than five, in usual cases, when the receiver is as large as the larger cylinder; losses which may be easily approximately estimated and allowed for in any case.*

In the next illustration from Mr. Porter's report, the natural form of the expansion-line, in the single cylinder, having the capacity here observed in the low-pressure engine, would be that shown by one or other of the two dotted lines, accordingly as the expansion approached more or less closely the hyperbolic form. The initial volume is AB , and the pressure as shown on the vertical scale; while the gradual loss of pressure with increase of volume is shown by the two scales as the line progresses toward the right to its terminal point at I . The deviation from the dotted line of the actual expansion-line between B and C illustrates the gain of weight and pressure due to the progressing re-evaporation of steam originally condensed in the cylinder at the opening of the steam-valve, and to the admission of the fluid into the colder cylinder. Here

* For exact expressions, see Sennett, Appendix; and Clarke's Manual, pp. 849 *et seq.*

expansion occurs from the initial pressure and volume at *B* down to the terminal point *C* in the high-pressure, and from *C* or *H* to *I* in the low-pressure, cylinder. The indicator-diagrams actually obtained are *ABCD* and *EFG*, the latter being the equivalent in the low-pressure cylinder of the card *HIJ*,

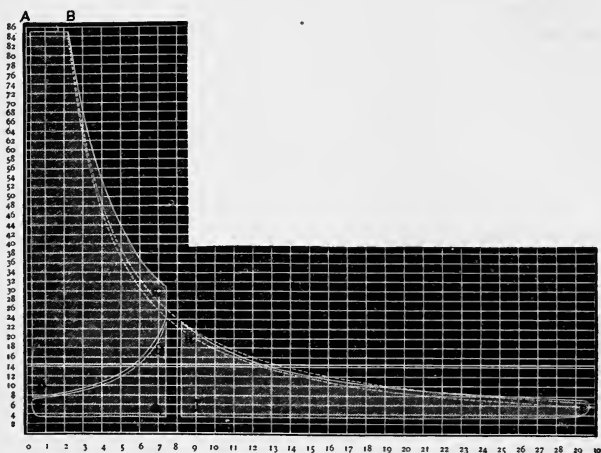


FIG. 65.—COMPOUND-ENGINE DIAGRAMS.

which would have been produced had the high-pressure cylinder been given sufficient length to permit the completion of the expansion in that cylinder. The variation of the full line, representing the real diagram, from the ideal dotted expansion-line is indicative of the fluctuations of pressure produced by the condensation and re-evaporations taking place as expansion progresses in the metallic chamber serving as working cylinder.

The succeeding figure illustrates the visible differences between the diagrams actually taken from the two cylinders of a compound engine—in this case a “Reynolds-Corliss”—and the ideal combined card.

This next diagram, from an engine of similar class with the

preceding and published by its designer, exhibits at once the method of reducing the actual indicator-diagrams to the combined form, and the variations from the ideal expansion-line due to imperfections of the engine as a work of human art. Pressures are measured in pounds on the square inch and volumes in cubic feet, actual capacities of cylinder being given. As shown on the diagram, about $3\frac{1}{2}$ cubic feet of steam enter

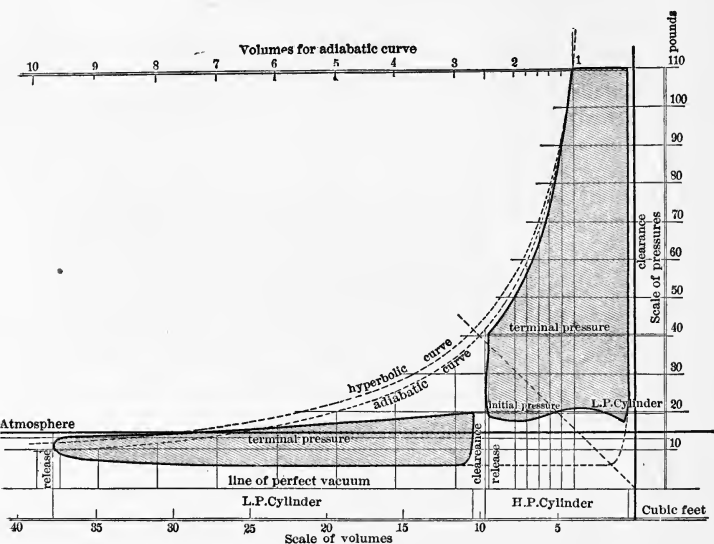


FIG. 66.—ACTUAL AND IDEAL DIAGRAMS; COMPOUND ENGINE.

the high-pressure cylinder, each stroke, at a pressure of 110 pounds per square inch above vacuum; it expands nearly adiabatically to $9\frac{1}{4}$ cubic feet, is then transferred to the low-pressure, dropping from the terminal pressure, 40 pounds in the high-pressure cylinder to 20 in the low-pressure, and then expanding in the latter down to about 12 pounds when it passes into the condenser, the back-pressure thus becoming not far from an average of 6 pounds. The two indicator-diagrams

are shown by the "hatched" spaces; the ideal diagram incloses both, its outline being the dotted lines. The very considerable space measuring the difference of the two areas is a gauge of the imperfection of the cycle. The departure of the actual line from the two ideal expansion curves, and the fact that the former lies within both the latter, indicate that the jacket does not supply heat enough to compensate the condensation of the expanding fluid; far less enough to retain its temperature constant or to continuously superheat it.

The discordant fluctuations of similar lines in the two indicator-diagrams exhibit the effect of non-synchronous motion of the two cylinders.

The accompanying illustration exhibits the proportions of the diagrams taken from a triple-expansion engine, drawn to common volume and pressure scales, and placed under the Mariotte line. The engine has cylinders having the ratios

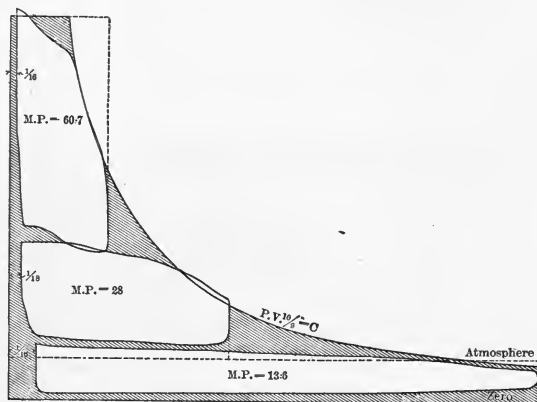


FIG. 67.—TRIPLE-EXPANSION DIAGRAM.

1 : 2.25 : 2.42, and the total ratio of expansion is 8, the cut-off in the several cylinders being set at 1.47, 1.3, 1.3. An advantage of this type of receiver engine, with its cranks making equal angles, is that the drop in pressure may be reduced to an

unimportant amount. Here steam-pressure is carried at 125 pounds by gauge; the efficiency 0.18, and the coal used 1.5 pounds per I. H. P. per hour.

62. Special Applications of the indicator are of peculiar interest to the engineer. Valve-adjustment is often performed, and should always be checked, by the aid of this instrument. The application of the indicator to the steam-chest, and the comparison of its readings with those of the ordinary diagrams and of the steam-gauge at the boiler, will often reveal defects in the steam-passages or valve-action otherwise difficult of detection; its use on the air-pumps of condensing engines and on the main pumps of pumping-engines similarly reveals anything objectionable in their construction and operation; and the motion being derived from the eccentric or the valve-mechanism when attached to the engine in the usual manner will permit a more minute examination of those phases of operation which are not easily studied on the common form of card. In some cases a continuous motion, derived from the crank-shaft, is adopted for this purpose.

In valve-adjustment, an inspection of the diagram shows the operation of the valve-mechanism as set. The necessity of adding lead or the reverse, of resetting the valves and eccentrics, is seen, and they are readjusted and diagrams again taken, these operations being repeated until the form of diagram desired is approximated as closely as is practicable.

In illustration of the forms of diagram obtained from the steam-chest and their interpretation, Mr. Porter gives those exhibited in the illustration.*

In these figures the steam-pressure fluctuates in the valve-chest with the draught upon it by the engine, rising to boiler-pressure after cut-off takes place, and falling below it more or less as the steam enters the valve-ports more or less rapidly. The extremities of the diagram correspond to the end of stroke of engine-piston; the points *c* to the points of cut-off at each stroke. The lower line shows the pressure in the chest during the interval up to the moment of action of the expansion-valve.

* The Richards Steam-engine Indicator, pp. 177-8.

Immediately afterward the pressure rises to boiler-pressure and there remains until the point is reached at which a drop of the upper line shows that the opposite end of the cylinder has begun to take steam. In *A* and *B* the pressure, when at its maximum, actually exceeds boiler-pressure, the surge of the mass in the steam-pipes and chest, when suddenly checked, causing a wave of abnormally high pressure. These were taken from Mr. Porter's engine at the Paris Exposition of 1867, when making 200 revolutions per minute.

In *C*, evidence is found of insufficient area of steam-pipe,

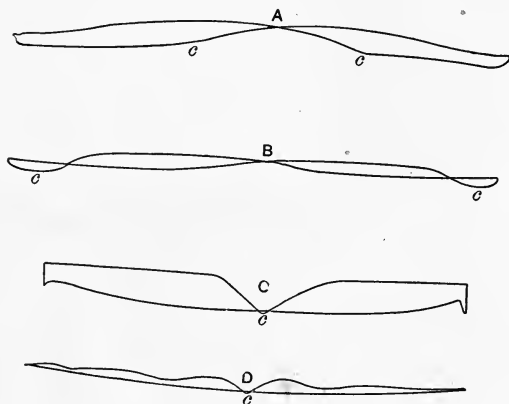


FIG. 68.—VALVE-CHEST DIAGRAMS.

resulting in the observed considerable fall of pressure when the engine takes steam at either extremity and the correspondingly large rise at the cut-off, *c*. The pipe having been enlarged, the card *D* was obtained; the sudden drop and continuous fall of pressure while the engine takes steam and the considerable loss of pressure, of power, and of efficiency indicated in *C* are in *D* all absent.

63. Pump Diagrams obtained by application of the instrument to the air-pump of a condensing engine are seen illustrated in the two succeeding figures, given by Mr. King, as

taken by him from the air-pumps of the U. S. S. "Powhatan," a paddle steamer of old type, having jet-condensers.*

With the first of the pair the engine was working as usual; in the second case the pump was taking in a large excess of air through the bilge-injection. The pump, in the first ex-

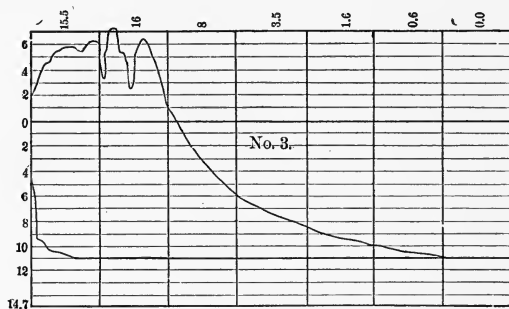


FIG. 69.—NORMAL AIR-PUMP CARD.

ample, draws the mingled air, vapor, and water from the condenser at a pressure about 4 pounds above a vacuum, throughout the induction-stroke, and on the return or educ-

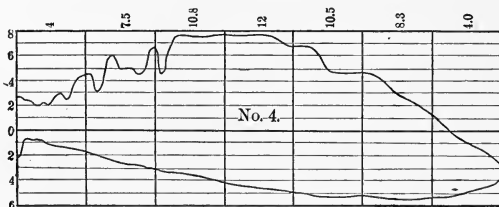


FIG. 70.—AIR-PUMP WITH EXCESS OF AIR.

tion-stroke they are compressed under a regularly increasing pressure to the point at which, the delivery-valve opening about 5 pounds above atmospheric pressure, the whole mass is discharged with fluctuations of the line due to rise and fall of the

* Practical Notes, pp. 56-60.

valve and irregular expulsion of air and vapor. The pressure then falls to about 2 pounds, and the end of the stroke is reached. Excess above the last observed pressure is due to friction in the discharge-passages and delivery-pipes. These diagrams are seen to be characteristically different from those obtained from the engine.

Pumps raising water, or any other incompressible fluid, should give a diagram like that here shown, as taken from the remarkable pumping-engine built by Mr. Corliss for the town of Pawtucket, R. I. This diagram is seen to be perfectly

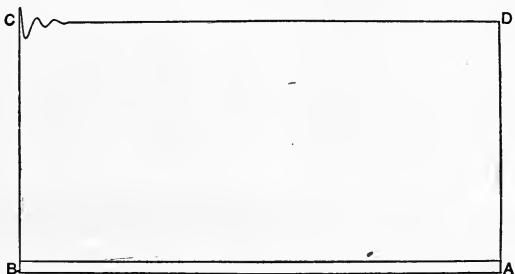


FIG 71.—PUMP DIAGRAM.

rectangular, the water entering from beginning to end of stroke *AB*, the pressure about five pounds below that of the atmosphere; the change to the delivery pressure, *BC*, a little above 100 pounds per square inch, taking place instantly on reversal, and the discharge, *CD*, occurring at uniform pressure. The slight disturbance at one corner, *C*, is due to a jar of the spring of the instrument.

Air-compression Diagrams exhibit this effect, as in the adjacent figures, showing a steam-engine and an air-compressor card taken from the Allen "positive valve-motion" compressor at the same time. The engine drives the compressor, and the work shown on the engine-card exceeds that of the compressor by the lost work of the apparatus. The engine is seen to have early admission, incomplete expansion, early release, and cushioning, all of which are practically to some extent required for

best effect; but the compressor takes in its charge throughout the induction-stroke, compresses steadily from the minimum to the maximum pressure, and has no observable variations from the ideal diagram such as are exhibited in the case of the steam-engine. This machine was running at 144 revolutions

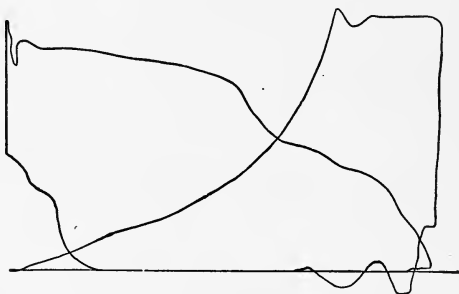


FIG. 72.—STEAM-ENGINE AND AIR-COMPRESSOR DIAGRAMS.

per minute when “indicated,” and this speed caused those fluctuations of the curve due to inertia in the instrument.

Diagrams taken with the motion of the paper derived from the main shaft are of the form seen in the next figure.

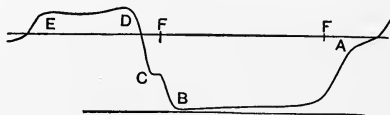


FIG. 73.—SHAFT DIAGRAM.

From *A* to *B* is the exhaust and vacuum line; from *B* to *C*, the cushioning; *C* to *D*, receiving-line; *D* to *E*, steam-line; *E* to *A*, expansion; *FF* are supposed to be the ends of the stroke.

Fig. 74 is the same diagram extended more nearly to its original length, the ends joined and then folded at *FF* so as to represent more nearly the usual diagram, but still preserving the peculiar length and proportion of lines. Diagrams taken in this manner expose more perfectly defective arrangements of valves; and a studious comparison of them in connection with

the usual form may often be useful in detecting defective sizes and proportions of valves, steam-passages, etc., and show very

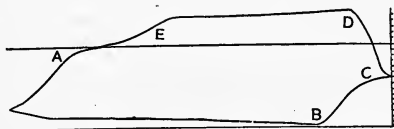


FIG. 74.—RECONSTRUCTED CARD.

correctly the proportion of time occupied in each different operation during the revolution of the engine.*

Attaching the line to the eccentric motion will give similarly useful diagrams.

64. Peculiar Forms of diagram are often met with, each invariably exemplifying some singular or abnormal arrangement of the engine.

| | | |
|-----------------------|--------|----------------------------------------|
| Steam in boilers..... | 9 lbs. | " Powhatan," Feb. 13, 1854. |
| Revolutions..... | 5 | Stb. cylinder bottom, working by hand. |
| Hot-well..... | 100° | |
| Throttle..... | 4 | |

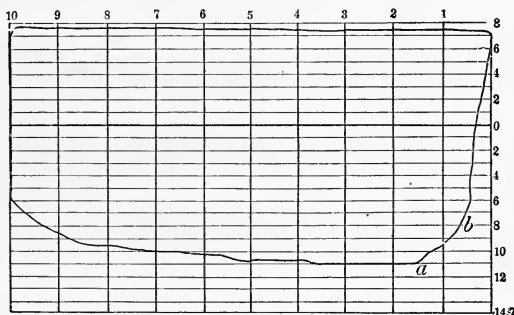


FIG. 75.—WORKING BY HAND.

Fig. 75, from King, is a diagram showing the operation of the valves while working by hand. This valve exhibits large

* Stillman on the Indicator, p. 31.

cushioning and steam lead, the exhaust-valve closing at *a*, and the steam-valve opening at *b*, so that the engine passed the centre against a pressure of $6\frac{1}{2}$ pounds above the atmosphere.

Fig. 76 is obtained from the same source as the last. In this case the engine was working as a non-condensing engine with a

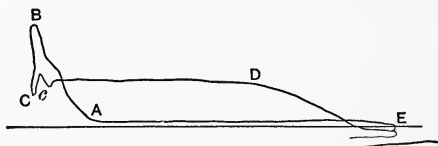


FIG. 76.—PECULIAR ADJUSTMENTS.

very low pressure of steam. The exhaust closes at *A* causing the pent-up steam to be compressed to *B*, where the steam-valve opens, and the pressure in the cylinder, being greater than that in the boiler, immediately falls to *C*. The hook at *C* is occasioned by the momentum of the indicator-piston. At *D* the cut-off closes, causing the steam to be expanded to *E*, below the atmosphere. At *E* the exhaust-valve opens and the pressure rises up equal to the back-pressure, causing the loop on that corner of the diagram.

Fig. 77 is a diagram from a non-condensing engine. The

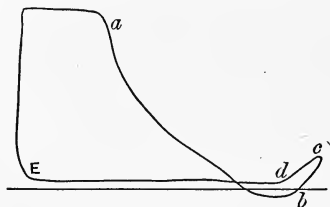


FIG. 77.—READMISSION.

pressure rises from *b* to *c*; but supposing the exhaust to open at *b*, there could be no reason why the pressure should rise beyond *d*, the amount of back-pressure on the opposite side of the piston; such a diagram could only be formed from a slide-valve engine, in this manner: Steam being admitted until the

piston arrived at *a*, the independent slide-valve cut off the steam; it was then expanded to the point *b*; at *b*—the steam-valve having deficient lap and lead, and thus being open—the cut-off valve opened, admitting fresh steam, which caused the line *bc* to be traced. At *c* the steam is shut off by the steam-valve itself and the exhaust opened; the pressure then falls to *d*, and the exhaust-line is traced.

Fig. 78 is an Otto gas-engine diagram, taken purposely, by

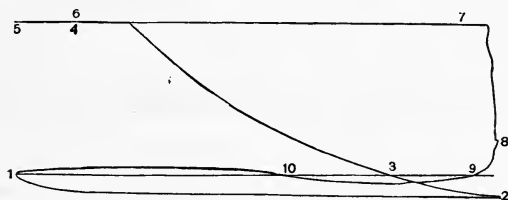


FIG. 78.—DIAGRAM TAKEN WITH LIGHT SPRING.

Messrs. Brooks and Stewart, with a light spring, in order to exhibit better the action of the machine at the more obscure points.* The induction-stroke begins at 1; the mixed gases are

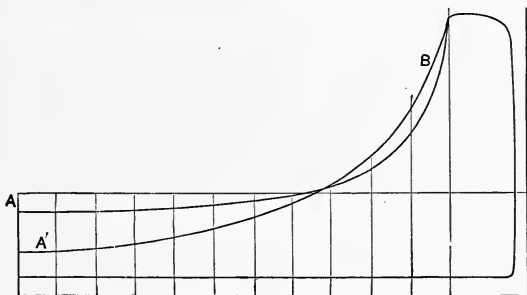


FIG. 79.—CYLINDER CONDENSATION.

taken in throughout the stroke to 2; compression occurs from 2, the pressure rising again to atmospheric at 3 and reaching the

* The Otto Gas-engine, Van Nostrand's Magazine, 1883.

limit of the spring at the line 4, 5, 6. The mixture is fired on the succeeding stroke, the pressure continuing above the limit of the spring to 7; then the exhaust-valve opens and the expulsion occurs, producing the line 7, 8, 9, 10, 1. The depressed part 9-10 may be due to inertia.

In the next example a steam-engine moving very slowly gives an expansion BA , which differs remarkably from the curve BA' of Marriotte, usually closely followed by good engines. The cause is now well known to be what is called internal or cylinder condensation, and re-evaporation—a phenomenon discussed fully elsewhere.

CHAPTER VI.

MEASUREMENT OF DIAGRAMS; COMPUTATIONS; APPARATUS AND METHODS.

65. The Apparatus and Methods of measurement of the power of the engine by means of indicator-diagrams are necessarily somewhat different with differences of purpose and of data desired. They include such as aid in the direct measurement of the diagrams, and also instruments employed to measure the speed of the engine and its fluctuations. Among the former is the planimeter; among the latter, speed-indicators, counters, and chronographs of various kinds. The methods of their use and of computations based on their work should always be such as will yield results of the greatest practicable exactness.

66. The Measurements taken in working of indicator-diagrams demand great care and accuracy. The figure to be measured is small; its bounding-lines often obscure and generally irregular; and the determination of its exact area, which is the usual problem, requires nice manipulations. An indicator-diagram represents the pressures, volumes, and work of the steam, or other fluid, at every instant throughout a single revolution of the engine, on *one* side of the piston. A pair of cards exhibits these quantities on both sides during one revolution. A series of such pairs exhibits the varying pressures and work of the engine at the several single revolutions to which they severally appertain. The average of the pressures shown on one card is the mean pressure for a single revolution on one side the piston; the average obtained from a series of diagrams gives a mean of the pressures, for the period covered, with a degree of approximation dependent upon the number of diagrams and the uniformity of action of the engine. By taking diagrams with sufficient frequency, any desired accuracy may be attained. In practice, they are often taken as seldom

as once an hour, and, on trials of importance, sometimes as often as every fifteen minutes. At sea, it is customary on naval vessels to take a set of diagrams once a day to be preserved in the log-book.

Since the diagram only gives the pressures, the other factors of work and of power must be determined otherwise. The indicated work of the engine at each stroke is the product of the net intensity of pressure on its piston by the volume traversed. The power is the work done by the engine in the unit of time; in British measures,

$$\text{H. P.} = 2 \frac{p \text{ plan}}{33000} :$$

where p is the average net effective pressure of the steam as shown by the indicator; l is the length of stroke; a is the effective area of piston; n is the number of revolutions per minute. Pressures are here, as usual, measured in pounds on the square inch, areas of piston in square inches, the stroke in feet, and work in foot-pounds per minute. Of these quantities, all but p are obtained by direct measurement and by observation. The pressure p is the one quantity obtained by the use of the indicator. The method of determination is to measure the area of the diagram, divide by its total length, and thus obtain in the quotient its mean altitude. This being multiplied by the scale of the spring and of the ordinates gives the mean pressure.

The *mean total pressure* is this quantity measured to the vacuum line of the card. The *mean effective pressure* is the mean pressure measured from the indicator-card, and that which represents the net pressure acting in the production of the indicated work. It is this pressure, usually, which is sought in making computations. The area of the diagram may be obtained in either of several ways.

The best method is by the use of the planimeter, which, with careful handling, should give the area to hundredths of a square inch. Divide the area by the length, and the result will be the height of a rectangle having equal area, and the average height of the actual diagram. Or:

Draw ten, or any other equidistant number, of lines, as in Fig. 80, perpendicular to the atmospheric line. The first and

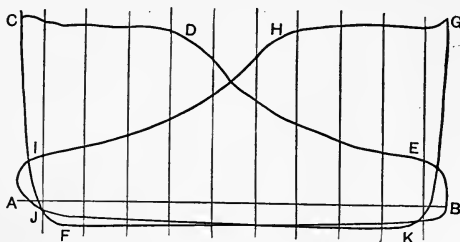


FIG. 80.—MEASURING THE DIAGRAM.

last half of the intermediate distance from the ends, and the height of each, represent the approximate height of the space which it marks. Measure the length of each ordinate, and divide the sum by the number of ordinates. Multiply the average length thus found by the scale of the spring, and the result is the mean effective pressure.

In case there is a loop, as in Fig. 81, caused by expanding below the back-pressure line, the engine being non-condensing,

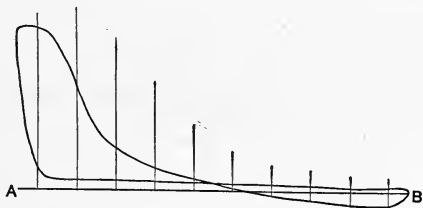


FIG. 81.—MEASURING DIAGRAMS.

the ordinates below are negative, and must be subtracted from the lengths of the ordinates above. Then, using the pressure so obtained, multiply the net area of the piston by the mean effective pressure (M. E. P.). Multiply this product by the distance through which the piston travels per minute, divide by 33,000, and then, as already seen,

$$\text{I. H. P.} = \frac{\text{Net area of piston} \times \text{M. E. P.} \times \text{revs. per minute} \times 2 \times \text{stroke.}}{33000.}$$

When there are a number of diagrams from the same engine, the calculations may be simplified by multiplying the area of the piston by twice the length of the stroke, and dividing the result by 33,000, thus grouping the constants, and thus obtain the "constant for the engine," the power developed at one revolution per minute with one pound mean effective pressure. If we multiply this constant by the number of revolutions and the mean effective pressure, the product will be the I. H. P. If the number of revolutions is constant, multiply the "*constant for the engine*" by the revolutions per minute. This gives "*horse-power constant*," or power developed per pound of effective pressure. Multiply this constant by the M. E. P., and the result will be the indicated horse-power.

A quick method of measurement of ordinates is to use a strip of paper, and mark, one after another, the lengths of ordinate on its edge, thus making the addition, and with a single final measurement. Lay the edge of the paper on the first line, mark off the distance, starting from the end of the paper; transfer the edge of the paper to the last line, add its length to the first measurement, and continue the addition for the intermediate divisions. Finally, measure, with the scale of the spring, the total length, and divide the result by *ten*.

A small adjustable set of parallel rods is often supplied by the makers with each pair of indicators, which may be used, as in the illustration, to lay off and describe the ten ordinates usually adopted in measuring the diagram. Or a scale dividing off ten parts in a space a trifle longer than the paper may be laid on the diagram diagonally, in such manner that its extremities will coincide with the ends of the diagram and the location of ten ordinates pointed off on the paper, between which the vertical measurements should be made. The use of a scale prepared like that seen in

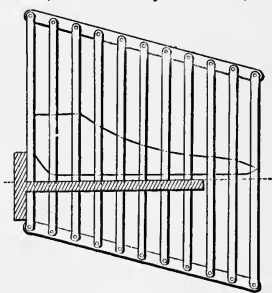


FIG. 82.—PARALLEL RODS.

the accompanying sketch, and which can be easily adjusted to any ordinary length of diagram, is still more convenient. Here the ordinates are correctly set, so as to give a half-space at each end to admit the measurement being made on the lines.

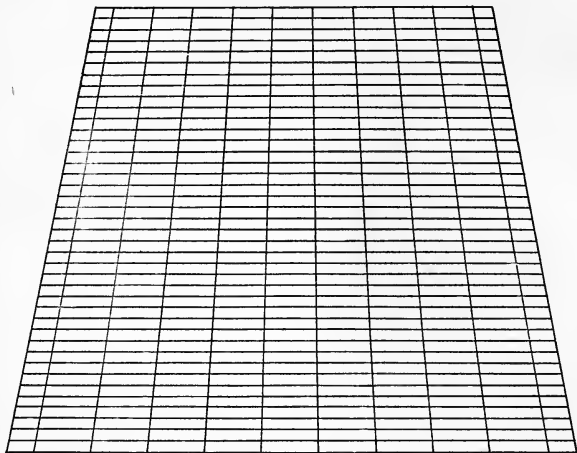


FIG. 83.—SPACING-SCALE.

A very rapid approximation to a correct volume of the mean pressure can be obtained by the expedient illustrated in Fig. 84.

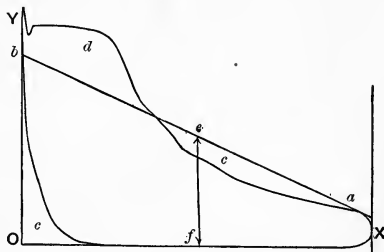


FIG. 84.—MEAN PRESSURE.

Ox being the back-pressure line, draw ab in such manner as to make the areas c and d between that line and the upper border of the card as nearly equal as possible. This can be quite closely judged by the eye. Then the ordinate, ef , drawn at the middle of the diagram gives the mean effective pressure.

The following, from Rankine, illustrates a good method of record and computation.* Ten ordinates are measured and the results for both cylinders of a compound engine are given.

COMPOUND-ENGINE DIAGRAMS.

| ORDINATES. | FIRST CYLINDER. | | SECOND CYLINDER. | |
|-----------------------------------------------------------------------------------------------------------|-----------------|---------|------------------|---------|
| | Top. | Bottom. | Top. | Bottom. |
| b_0 | 27 | 36 | 16.0 | 12.4 |
| b_{10} | 13 | 12 | 2.0 | 3.8 |
| Sum | 40 | 48 | 18.0 | 16.2 |
| Half sum | 20 | 24 | 9.0 | 8.1 |
| b_1 | 83 | 97 | 10.5 | 10.8 |
| b_2 | 91 | 96 | 8.5 | 9.0 |
| b_3 | 91 | 84 | 7.5 | 8.0 |
| b_4 | 64 | 64 | 7.0 | 7.1 |
| b_5 | 57 | 57 | 6.6 | 6.7 |
| b_6 | 53 | 46 | 6.2 | 6.0 |
| b_7 | 42 | 40 | 6.0 | 5.6 |
| b_8 | 35 | 32 | 5.1 | 5.4 |
| b_9 | 22 | 22 | 4.5 | 5.0 |
| Sum | 558 | 562 | 70.9 | 71.7 |
| Sum $\div 10 =$ M. E. P. | 55.8 | 56.2 | 7.09 | 7.17 |
| Mean, top and bottom. | 56.0 | | 7.13 | |
| \times area of piston, sq. ins. | 345 | | 1380 | |
| Mean effort, in lbs. | 19320 | | 9839.4 | |
| \times stroke, in feet, $2\frac{1}{2} \times$ revol- utions per minute, $52\frac{1}{2} \times 2 =$ } | 262.5 | | 262.5 | |
| Ft.-lbs. per minute. | 5071500 | | 2582842.5 | |
| Total | 7654342.5 | | | |
| $\div 33000 =$ | 232 I. H. P. | | | |

* Steam-engine, p. 51.

These mean pressures are found by a process which may be algebraically represented as follows:

Divide the length of the diagram into any convenient number, n , of equal parts, and measure the ordinates at the two ends and the $n - 1$ points of division; so that ordinates are measured from $n + 1$ equidistant points.

Let p_0 be the first, p_n the last, and p_1, p_2 , etc., the intermediate ordinates of the upper curve; let p'_0 be the first, p'_n the last, and p'_1, p'_2 , etc., the intermediate ordinates of the lower curve; let p_m denote the mean forward pressure, p'_m the mean back pressure, and $p_m - p'_m$ the mean effective pressure. Then

$$p_m = \frac{1}{n} \left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.} \right)$$

$$p'_m = \frac{1}{n} \left(\frac{p'_0 + p'_n}{2} + p'_1 + p'_2 + \text{etc.} \right);$$

$$p_m - p'_m = \frac{1}{n} \left(\frac{p_0 + p_n}{2} + p_1 + p_2 + \text{etc.} - \frac{p'_0 + p'_n}{2} - p'_1 - p'_2 - \text{etc.} \right).$$

The mean effective pressure may be computed at once by measuring a series of equidistant breadths of the diagram the mean of which breadths represents the mean effective pressure. That is, let b_0 be the first, b_n the last, and b_1, b_2 , etc., the intermediate breadths.

Then

$$p_m - p'_m = \frac{1}{n} \left(\frac{b_0 + b_n}{2} + b_1 + b_2 + \text{etc.} \right).$$

The effective energy exerted by the steam on the piston during each revolution is twice the product of the mean effective pressure, the area of the piston, and the length of stroke, or

$$2(p_m - p'_m)As;$$

and if N be the number of double strokes in a minute, the indicated power in foot-pounds per minute is

$$2(p_m - p'_m)ANs;$$

from which the indicated horse-power is found by dividing by 33,000.

The presence of the piston-rod or of a "half-trunk" on one side the piston produces a difference of areas which, in the latter case, is of considerable magnitude. Where measured separately, if A_1 and A_2 are the two areas, the power is

$$\text{I. H. P.} = \frac{(p_1 A_1 + p_2 A_2) l n}{33000},$$

l and n being the length of strokes of piston and the number per minute.

67. Planimeters, plane-area measuring instruments, mechanical integrators, as they are variously called, furnish the best known means of measuring the area of the indicator-diagram. By their use the work can be done by an expert hand with great rapidity and with marvellous accuracy. Liability to error is exceedingly small, and the magnitude of the probable error is quite inappreciable in the ordinary work of the engineer*—sometimes as little as one part in above ten thousand. Errors exceeding one tenth of one per cent are usually due to inexperience of the operator. The best known instrument is that of Amsler; that of Coradi,† of similarly general application, and that of Coffin,‡ designed especially for the measurement of indicator-diagrams, are also in common use, the two former mainly in Europe, the latter in the United States. They commonly operate by the combined sliding and rolling motion of a small measuring wheel which has a total rotation proportional to the area enclosed by the figure the periphery of which it traverses.§ The integrating wheel, or roller, is best made of steel. A vernier on the instrument en-

* "Ueber die Genauigkeit der Planimeter;" Professor Lorber; *Oesterreiche Zeitschrift für Berg- und Hüttenwesen*; vol. xxxi; p. 22.

† Ueber das Roll Planimeter von Coradi; Franz Lorber; *Zeitschrift des Oesterr. Ing. & Arch. Verein*; vol. xxxvi; p. 135.

‡ Barrus on the Indicator, p. 61.

§ Bramwell on the Amsler Planimeter. Report Brit. Association, 1872, p. 404. Shaw on Mechanical Integrators; Proc. Brit. Inst. C. E., 1884-5, No. 2063.

ables the readings of the motion of the roller to be taken with great accuracy, and repeated measurements may be made to eliminate errors less in amount than the finest readings given. A very simple modification of the Amsler Planimeter, designed by Mr. J. W. See, is made especially for indicator work.

The following are the maker's instructions, in detail, for using the most recent form of Amsler planimeter:

1. Adjust the screw-centres upon which the index-roller *D* revolves, so that the roller works freely, and does not touch the vernier. The same care must also be taken with the centre-pin *C*. Oil the screw-centres now and then. Care should be

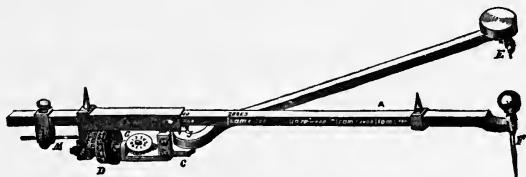


FIG. 85.—AMSLER PLANIMETER.

taken to prevent the tube *B*, the tracer *F*, and the point *E* from being bent, and also to see that the barrel *D* is kept uninjured.

2. To find the area of any figure, set the roller *D* and the counting-wheel *G* to zero; the square rod *A* must be pushed into the tube *B*, and the line on *A* marked 1 sq. dem., or 0.1 sq. ft., etc., must come even with the small line on the bevelled part of the tube *B*; when this is done, place the instrument on the paper, and see that the roller *D*, the tracing-point *F*, and the needle-point *E* touch the paper. Press the point *E* slightly into the paper, and put the small weight on the hole over the point; the instrument is then ready for work.

3. Take any point *P* on the outline of the figure about to be measured, set the tracing-point *F* to that point, and when it is marked, read off the index-roller *D* and counting-wheel *G*. For example, suppose the counting-wheel shows 2, the roller 91, and the vernier 5, the number will be 291.5. Follow the

outline of the figure with the point *F* as accurately as possible to the right, until you come to the starting-point. Straight lines can be followed along a ruler; then read off the numbers on wheel and roller; say it is the second time 476.7.

4. When these two numbers are obtained, there are two cases to be observed:

1st. If the point *E* is outside the figure, subtract the first reading 291.5 from the second 476.7, the remainder is 185.2, which shows that the area contains 185.2 units. Of course the units depend entirely on the regulation of the bar *A*, if they are 0.1 sq. ft. we have $185.2 \times 0.1 = 18.52$ sq. feet, as the area of the figure measured on the paper.

The rule therefore is, when the point *E* is outside, multiply the difference of the two readings by the number on the bar to the right of the corresponding division.

2d. When the point *E* is inside the figure, before making the subtraction, the number engraved on the top of bar *A*, above the corresponding line of division, must be added to the second reading. In this instance, suppose the number on top of bar *A* is 20.985, the second reading is 4.767, the calculation would be as under:

| | | |
|---------------------------------|---|--------|
| Second reading | = | 4.767 |
| Number over 0.1 sq. ft. | = | 20.985 |
| | | 25.752 |
| Deduct first reading | = | 2.915 |
| | | 22.837 |

The area is therefore 22.837 units or $22.837 \times 0.1 = 2.2837$ square feet. It is of no consequence whether the roller is inside or outside the figure, provided it is on the same level.

In using the Coffin planimeter, the diagram is set with the atmospheric line parallel with the lower edge of the clamp *C*, and the end even with the perpendicular edge. The clamp *K* is moved up to the other end. The block *Q* being introduced into the groove *I*, the tracer is set upon the point *D*, where the edge of the clamp *L* touches the figure. The wheel is

turned so as to bring the reading to zero, and the tracer is then moved over the line of the diagram in the direction of the motion of the hands of a watch. The tracer is then moved along the edge of the clamp till the reading is again made to

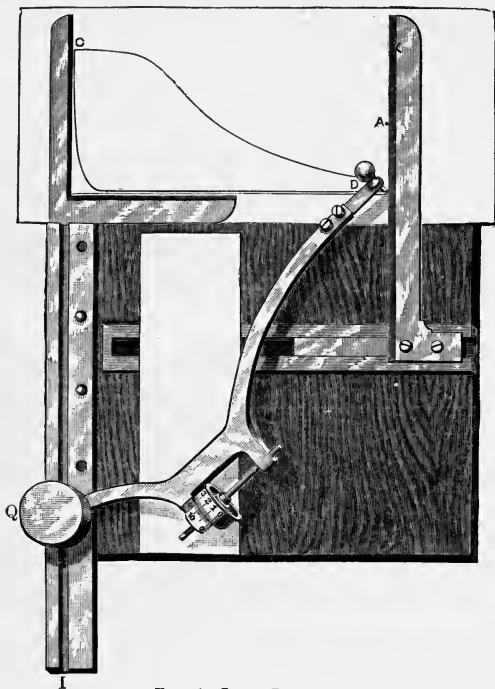


FIG. 86.—COFFIN PLANIMETER.

zero. The distance of *A* from *D*, now measured on the scale of the spring, is the mean effective pressure. The reading on the wheel is the area of the diagram in square inches. From this area the mean effective pressure may be also computed,

by multiplying it by the quotient obtained by dividing the number of the spring by the length of the diagram in inches.*

This measurement may be made and computations computed on forty or fifty diagrams per hour by an expert computer using this instrument, obtaining the value of the mean effective pressure, inserting it in the formula already given, and computing the indicated horse-power.

Mr. Lea has made an interesting modifier of the ordinary indicator by the substitution of a planimeter for the pencil-motion,—either permanently or temporarily, as may be desired,—thus at once getting a measure and registry of the work done.†

68. Tachometers, Speed-indicators, Chronographs, and Counters are instruments of various kinds and classes used by the engineer in determining the speed of the engine, the second of the essential factors obtained by observation in measuring its power. Of these instruments, the counters and many speed-indicators give the exact and positive measures of the engine-speed required by the observer as data. They mechanically register the revolutions of the machine one by one, and give either the totals or the differences for selected intervals; or, more usually, they work continuously, and these totals or differences are read off at regular intervals by the observer and recorded in the log, thus giving the means of obtaining the average speed of engine throughout the trial. When indicator-diagrams are taken, the speed of rotation of the engine is taken as nearly simultaneously as possible. This measurement is commonly taken with one of the small hand speed-indicators, the spindle of which is applied to the centre in the end of the main shaft and there held a quarter or half minute, a full minute, or more, as less or greater accuracy is desired and as the speed of the engine is greater or less. Very exact measurement is usually demanded for purposes of computation.

The “tachometers” are a class of instruments which exhibit

* Barrus, p. 63.

† Mechanics; Jan. 20, 1883; p. 39.

on a dial the speed of rotation of the shafts by which they are driven. They are not generally relied upon to give exact readings; but their closely approximate indications check the hand-counter record at any moment, as the hourly or daily readings of the mechanically registering counters permanently attached to the engine check the averages of that record. The tachometers, Fig. 87, are actuated by pulleys or gearing, and are



FIG. 87.—TACHOMETER.

designed to indicate the number of revolutions performed per minute by shafting, by a pointer travelling over a graduated dial. In the cylindrical case rotate two suspended weights or pendulums, connected together by a strong flat coiled spring. The purpose of this spring is to counterbalance the centrifugal force of the pendulums. The deviations of the pendulums are communicated by a rod to clockwork in the case behind the dial, and produce corresponding deflections of the pointer. These instruments possess the advantages of exhibiting to the eye the momentary fluctuations of speed which cannot be thus shown by the revolution-counters. In some cases, by the addition

of a recording mechanism and a roll of paper to receive the record, the tachometer is converted into a "tachograph," and it is in this form often attached to engines or other machines to supply a constant and permanent record of their operations. For experimental purposes the paper is driven at comparatively high speed, as high as an inch (2.5 cm.) per minute, the more common speed being one half or one quarter that velocity. When used on the locomotive engine, it is customary to mark the dial in miles per hour as well as revolutions per minute. Various forms of the instrument are made for the various purposes of engineering, and are applicable to all speeds—from that of the slowest engine to that of the fastest electric machinery. The Edson speed-recording instrument, and the Moss crop and other familiar apparatus, are of this class.

The modern *Speed-recorder* is an instrument, Fig. 88, which registers the fluctuations of speed of the engine, or of other machinery, on a travelling strip of paper actuated by a clock. The variations of velocity are produced by the movement of a revolving pendulum, like the Watt governor, which moves a pencil across the line of motion. The curve thus traced is a record of the whole history of the time of observation. A widely serrated line shows great irregularity of speed; the less and the closer the serrations, the better the speed; the rise of the line above the mean indicates a steady acceleration; a fall means retardation; a wide, even, band of oscillations shows a light wheel; a narrow band along the correct mean indicates a good balance-wheel; a gradual fall may indicate change of steam-pressure, and sudden variation of location of the record line or band usually indicates fluctuation of the load, and its extent of fluctuation the effectiveness of the system of regulation. The kinds of unsteadiness due to changing load and pressure and to inefficiency of regulation are easily distinguished, and the various causes of variation of speed may usually be easily traced out and remedied.

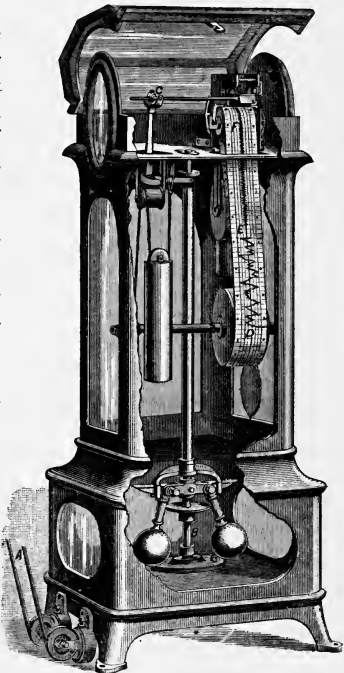


FIG. 88.—THE MOSSCROP RECORDER.

Chronographs, such as are employed by the physicist, may be used when it is desired to determine the method and extent of variation of speed during any single revolution of the engine,

or part of the stroke, as in investigating the effect of the varying pressure on the piston and the torsional moment at the shaft; conditions which can only be studied experimentally by the use of apparatus of extraordinary delicacy and quickness of action and record.

The chronograph was first applied to measure the variations of velocity of the steam-engine by a committee of the British Association in 1843-4, in their determinations of the speeds of piston of the Cornish engine.* It has been applied by Mr. Woodbury, as early as 1873, to pumping-engines, determining the fluctuations of velocity of fly-wheel, and, later, by Mr. Eckart in observing similar fluctuations of pump-rod speed, at great depths in the Comstock lode in Nevada.† The latter found it practicable to use the instrument at speeds of from 80 feet per minute up to 1400 feet. Messrs. Dix and Mack, under the supervision of the Author, applied the same instrument, still later, to the "high-speed" engine, making 250 revolutions per minute.

The following are Mr. Woodbury's records from three engines:‡

| Portion of Revolution. | Lowell Pumping-engine. | Lynn Pumping-engine. | | Horizontal Engine. |
|------------------------|------------------------------|----------------------|-------|--------------------|
| | Revolutions per Minute. | | | |
| | 13.26 | 18.61 | 13.90 | 19.39 |
| | Velocity in Feet per Second. | | | |
| .00 | 6.42 | 9.80 | 7.13 | 10.16 |
| .04 | 6.46 | 9.82 | 7.27 | 10.62 |
| .08 | 6.54 | 9.92 | 7.53 | 10.84 |
| .12 | 6.68 | 10.08 | 7.72 | 10.96 |
| .16 | 6.84 | 10.14 | 7.75 | 10.90 |
| .20 | 6.96 | 10.16 | 7.70 | 10.72 |
| .24 | 7.06 | 10.10 | 7.53 | 10.42 |
| .28 | 7.10 | 9.96 | 7.33 | 10.04 |
| .32 | 7.06 | 9.70 | 7.02 | 9.58 |
| .36 | 7.02 | 9.43 | 6.60 | 9.25 |
| .40 | 6.92 | 9.23 | 6.20 | 9.14 |
| .44 | 6.82 | 9.08 | 5.87 | 9.27 |
| .48 | 6.77 | 9.00 | 5.73 | 9.66 |
| .50 | 6.75 | 8.97 | 5.71 | 9.92 |

* Trans. B. A. A. S., 1844.

† Trans. Am. Soc. M. E.; Vol. III.; 1882.

‡ Ibid.

The curves here shown represent the motion of the fly-wheel of the Lynn engine.

By the use of the ordinary form of physicist's chronograph, or a slightly modified instrument, the speed of engine, and its variations, are measured, not only stroke by stroke, but even from point to point in the single revolution of the engine. This is a matter of importance in the application of engines, and especially if at low speed, to the driving of dynamo-electric

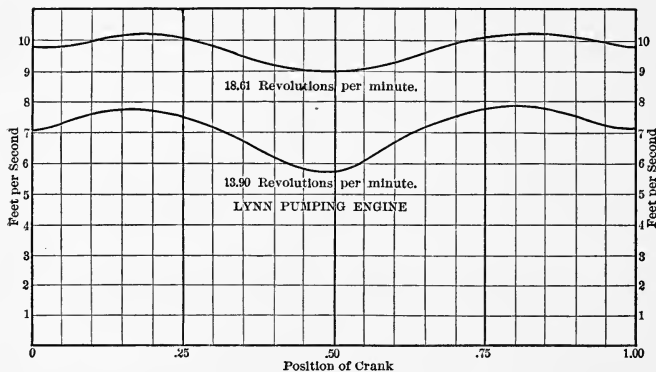


FIG. 89.—MOTION OF FLY-WHEEL.

machinery, where variations of speed, however limited as to time, are seriously objectionable. The frontispiece exhibits the method of attachment of the instrument to a "high-speed," direct-acting, horizontal engine of common type. *A* is the steam-cylinder; *B*, the engine-frame; *C*, the end of the main shaft; *D*, the balance-wheel; *E*, the brake-pulley, with strap *F*, and scale weighing the turning effort at *G*. On the extremity of the shaft, a coupling, *H*, is attached which drives the chronograph, *I*, through a slender rod seen connecting them. The revolving cylinder, on which the paper is smoothly stretched, to receive the record, is seen at *J*, and the stylus or pen is at *K*. The whole is mounted on a firm support as *L*.

When in operation, the cylinder is turned by the engine,

instead of its usual motor apparatus, and the pen, slowly traversing the cylinder, produces a closely compressed helix. At regular intervals, a circuit is made and broken by the standard clock or other timing instrument, and the line is given a V-shaped jog which marks the time-interval on the cylinder. The adjustment should be such that, at normal speed, these breaks should occur at precisely the same points in the circumference of the chronograph cylinder at each of its revolutions or at each tenth or other fraction of a revolution, as may be determined upon. Any acceleration or retardation will then be exhibited by the production of the break in advance of, or behind, its normal position. In the first case all such breaks fall into straight lines along the cylinder, parallel to the axis; in the latter case they will fall into regular helical, or curvilinear, or irregular, lines, accordingly as the acceleration or retardation is uniform, smoothly variable, or irregular. The inclination of the lines, or of the tangents to the curves so produced, to the axis is thus a measure of the change of speed.

Thus, if

C = the circumference of the record cylinder;

d = distance traversed by the pen per revolution;

n = number of revolutions of engine per minute;

n' = revolutions of the chronograph;

$$c = \frac{n}{n'};$$

θ = angle made by the line produced as above with the axis,—

then we have

$$\frac{Cn}{60c} = \frac{C \pm d \tan \theta}{2n'}, \quad \text{and} \quad n = \frac{30c}{n'} \left(1 \pm \frac{d}{C} \tan \theta \right);$$

using the positive and negative signs for acceleration and for retardation, respectively.

$$\text{When } \theta = 0, \quad n = \frac{30c}{n'}.$$

For a case of actual work, $C = 21.84$; $d = 0.0833 +$; $r = 285$; $n = 30$; and

$$n = 285 \left(1 \pm \frac{d}{C} \tan \theta \right);$$

and, making a variation of 1° , the angular deviation would become $\tan^{-1} \theta' = 42^\circ 36'$; since

$$n - 285 = \frac{nd}{C} \tan \theta' = 1,$$

and

$$\tan \theta' = \frac{C}{nd} = \frac{0.833+}{21.84}; \tan^{-1} \theta' = 42^\circ 36'.$$

It often requires very nice adjustment to secure sufficiently perfect arrangement of speeds to give a good line for the normal operation of the engine; especially as the sensitiveness of the instrument increases with decreasing values of the angle θ .

The following is a set of data thus obtained in the course of a trial of a good type of engine, considerably out of adjustment :

| Observation. | θ | $n_1 - n_2$ | n_2 | <i>D. H. P.</i> |
|--------------|-----------------------|-------------|--------|-----------------|
| 1 | $+ 75^\circ 51' 80''$ | $+ 4.35$ | 289.35 | 0 |
| 2 | $- 72^\circ 14' 17''$ | $- 3.55$ | 281.45 | 4.22 |
| 3 | $- 78^\circ 32' 58''$ | $- 4.54$ | 280.46 | 7.01 |
| 4 | $- 79^\circ 48' 46''$ | $- 6.05$ | 278.95 | 9.77 |
| 5 | $- 80^\circ 43' 54''$ | $- 6.63$ | 278.37 | 12.53 |
| 6 | $- 85^\circ 18' 45''$ | $- 13.26$ | 271.74 | 14.95 |
| 7 | $- 85^\circ 58' 47''$ | $- 15.47$ | 268.53 | 17.45 |

The next engraving exhibits the chronograph as used by Mr. Eckart. The reference-letters are as follow :

CC—Cast-iron base-plate, covered with sheet-brass, upon which the mechanism is secured.

B—Metal frame containing gearing for driving drum *A* and escapement-wheel *b*; motion communicated by means of adjustable weights *D*.

AA—Light brass drum, accurately balanced, revolving on friction-rollers 8, 8, at both ends.

ff—Parallel guide-bars upon which the tracing-point *h*, and its carriage travel back and forth, receiving motion in one direction from the engine or other moving parts through the cord *P*, passing through the bars *f*, and attached to the tracing carriage; the return motion is derived from a coiled spring in the spring drum *C*.

ee—Small electro-magnets on tracing carriage for raising the tracing-point *h*, off the paper and replacing it at any desired point to be especially observed.

d—Electro-magnets on separate carriage *kk*, adjustable on parallel bars *f*, operating the steel tracing-point *g*, at-

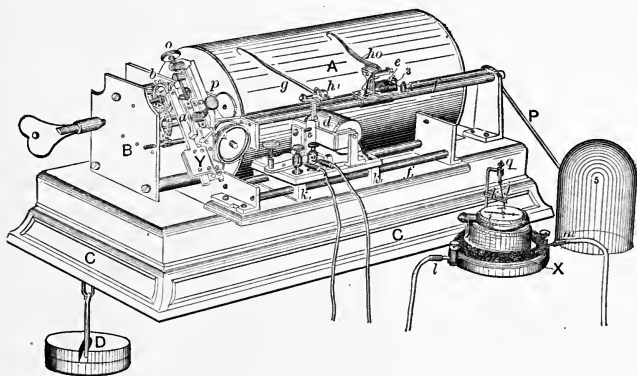


FIG. 90.—THE CHRONOGRAPH.

tached to the armature of *d*, for the purpose of recording seconds on the margin of the paper or at other parts of same as required.

i—Chronoscope or watch supported on frame *x*, the second-hand of which swings the light platinum wire *J*, breaking contact with the insulated wire *k*, thereby breaking circuit with *d* and recording seconds through the tracing-point *g* on the paper.

q—Adjusting screw for the wire *J*.

a—Steel spring of escapement. This spring is securely

clamped in *Y*, its flexibility being controlled to a certain extent by means of the thumb-screws *o* and *p*.

This instrument was found to give with great exactness the fluctuations of piston-speed in a pumping engine at 80 feet per minute, and for a hoisting engine at 1400 feet. It is thus peculiarly well adapted for pit-work.

Instead of, as is usual, employing a clock pendulum to mark time and give the velocity-curve of the engine, a portable time-keeper is here used. This is the common form of "timer," designed especially for timing in trials of speed of vessels, or on the race-course. The hand of the instrument, revolving once per second, breaks the circuit, and the stylus or pen *g* is caused to mark the interval. The stylus or tracing-point barely touches the lamp-black, being counterbalanced in such manner as to remove the coating without bearing perceptibly upon the paper, producing a fine white line on the black surface. In fitting the paper, it is cut slightly longer than the circumference which it is to cover, and lapping the edges and gluing them together, the lap is carefully sandpapered to as nearly as possible uniform thickness at the joint and elsewhere. The surface is then smoked, and is ready for use.

The Hand Speed-indicator is made in various forms. One which the Author has found very satisfactory is shown in the illustration. It answers equally well in whichever direction the engine may turn, is convenient in use, and gives reliable results. The Author has often found a different shape of point more certain to hold, and has often flattened the point and given it a semicircular sharp-edged termination, to obtain a more secure hold on the centre of the shaft.

The usual method of counting the revolutions of the engine, by means of the hand speed-indicator or the registering "counter" attached to the machine, gives the mean speed for a certain time—as for a minute—for which the count is taken. The use of the chronograph in the manner just indicated gives a measure of the *rate* or speed for the instant, for each revolution. To ascertain the rate during successive *portions* of the revolution, the method of Woodbury or of Eckart may be adopted.

These processes may all be used—properly fitting up the chronograph for the purpose; but a much less costly, more

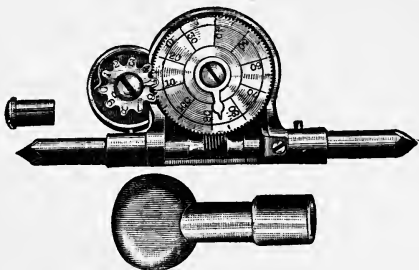


FIG. 91.—DOUBLE POCKET SPEED-INDICATOR.

convenient, and simpler method may be employed, the measuring instrument being the ordinary tuning-fork or "timing-fork."

The Timing-fork used in timing engines, and in measuring speeds and speed-variations during a single revolution of the engine, may be of any convenient size, but preferably one of slow vibration and low note. The fastest engines ordinarily make about one revolution in the fifth of a second; but very small engines, and especially those used in electrical and high-speed torpedo-boat work, sometimes make a revolution in the tenth of a second. A standard C fork making 256 vibrations per second would thus space off the engine-cycle into from 25 to 50 parts with the fastest of small engines, or into 256 parts if the engine makes 60 revolutions per minute. The normal fork, for concert pitch, at the Conservatoire de Musique, Paris, makes 870 vibrations for standard *a*, treble staff, corresponding to 261 vibrations for C. This would similarly measure speeds at intervals of 87 or 174 parts for the fastest engines, or 870 parts at 60 revolutions.

The usual method of application is that of Mons. Duhamel, who covers an accurately turned cylinder with a sheet of paper having a smooth and firm surface, uniformly covered with lamp-black, and permits a fork mounted on a firm stationary support to record its motions in a sinuous curve on the paper, as the cylinder is regularly turned beneath the point of a light stylus fixed on the end of the fork. The rate of the fork being

known, and the number of sinuosities of the line being counted for any specified period, the time becomes known. If the line be marked at each revolution or specified part of a revolution of the engine, by any convenient automatic system, the velocities become known for each of those periods. The cylinder carrying the paper should be caused to traverse longitudinally by the action of a screw of conveniently chosen pitch, as is illustrated by the recording mechanism of the Scott phonograph.* The record on the smoked paper being made, it may be saved and rendered permanent by dipping it in an alcoholic solution of shellac, or of sandarack or other gum.

In engine-testing the following method has been found to be very satisfactory: A toothed wheel or disk is mounted on the end of the main shaft, the number of teeth being determined by the degree of accuracy sought, as 36 to give measures for each ten degrees in the revolution of the crank and shaft, or 72 for five-degree intervals. These teeth constituted the circuit-breaking apparatus for a small battery, the current from which was the primary for a small induction-coil, the induced current being caused to pass through the stylus and to the paper-cylinder, each spark breaking through the smoked surface and leaving its mark, and the space and time between successive points thus made giving a measure of the speed of the engine in that ten-degree interval. Some care is necessary to get a good form of stylus for this work. The sketch shows that adopted by Messrs. Dix and Mack for their investigation of this subject. A light metal frame, *AA*, carries the very fine and light needle, or stylus, *B*, of which the point *P* is smoothly rounded so as not to tear the paper, and which is guided by an opening, *C*, and held up to a gentle contact by the spring of *D*. A small screw, *S*, holds the whole firmly in place on the end *F* of the fork.

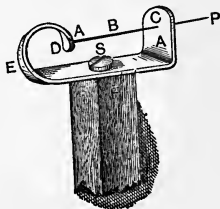


FIG. 92.
STYLUS FOR TIMING-FORK.

* See Ganot's Physics, § 246.

The next figure shows an improvised and simple, but effective, method of moving the fork. *B* is the engine-frame; *C* is

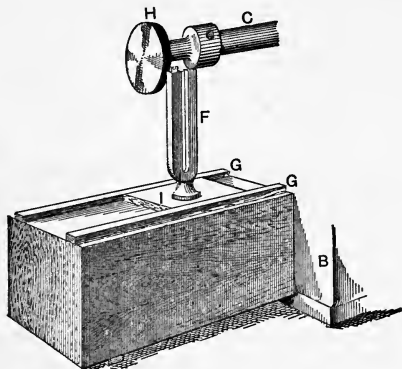


FIG. 93.—MOUNTING THE TIMING-FORK.

the crank-shaft; *F* the fork, mounted on a platform, *GG*, in such manner that it may be smoothly and steadily traversed

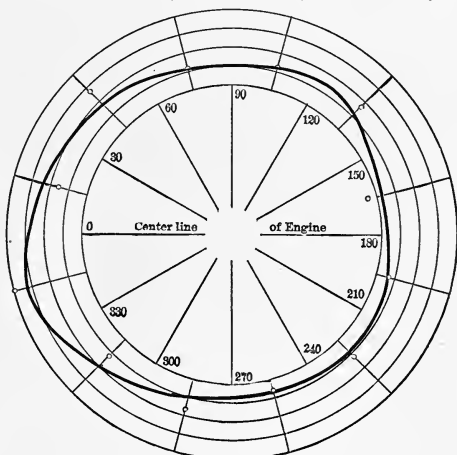


FIG 94.—VARIATION OF ROTATION.

before the smoked cylindrical surface *H* by sliding its base-piece, *I*, between the guides.

Below are the data thus secured by test of an engine making about 285 revolutions per minute :

| Angle. | Vibrations. | Variation. Per Cent. | Angle. | Vibrations. | Variation. Per Cent. |
|---------|-------------|-------------------------|-----------|-------------|-------------------------|
| 0°-30° | 8.7 | -2.3 | 180°-210° | 8.8 | -3.4 |
| 30-60 | 8.6 | -1.2 | 210-240 | 8.75 | -2.9 |
| 60-90 | 8.75 | -2.9 | 240-270 | 8.8 | -3.4 |
| 90-120 | 8.75 | -2.9 | 270-300 | 8.7 | -2.3 |
| 120-150 | 8.7 | -2.3 | 300-330 | 8.75 | -2.9 |
| 150-180 | 8.9 | -4.6 | 330-360° | 8.5 | 0 = normal. |

Plotting these data, the accompanying figure is obtained, and a comparison of the curve with that representing the varying accelerating moments acting on the engine-shaft; the two are found to accord—as should be expected. Radii here measure velocities. The connecting-rod was here six cranks in length, the ratio of expansion about four.

The accompanying illustration exhibits the mounting of the

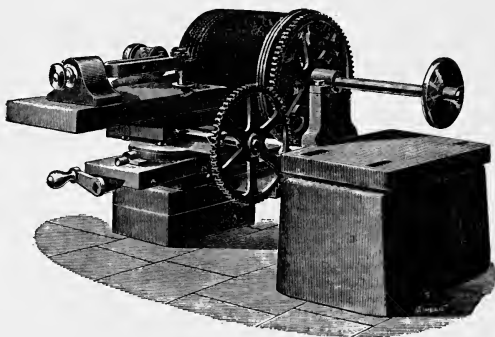


FIG. 95.—TIMING-FORK.

timing-fork as devised by Mr. Ransom.* The timing-fork, kept in motion by an electric current, is mounted on a rest driven by a screw, parallel to the axis of the paper-cylinder.

* Journal of the Society of Arts, Feb. 15, 1889, p. 243.

The operation of the instrument is the same as in the cases already described. The record obtained is similar, and may be illustrated by that given, full size, in the next figure, which represents a mean speed of 141 revolutions per minute.

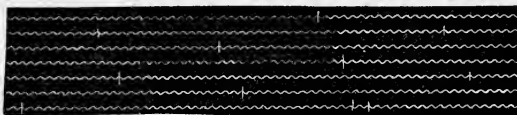


FIG. 96.—SPEED RECORD.

Good regulation is usually considered to imply:

(1) Uniform rotation; meaning minimum variation of angular motion during the revolution of the crank-shaft.

This variation depends, for its amount, upon the simultaneous variations of effort and resistance, and upon the magnitude of the regulating mass, the fly-wheel.

(2) The speed of engine, revolution by revolution, should be very nearly constant.

This variation should not usually be allowed to exceed two, and is often less than one, per cent.

(3) The mean engine-speed should remain constant over the whole period of its operation.

(4) The mean speed of the engine should be precisely that at which it is intended to be operated, irrespective of variation of load or of steam-pressure.

The Computation of Power, or of the work of the steam on the piston, is possible whenever, the dimensions of the engine being known, the mean pressure on the piston and its mean velocity have been measured for the required period. The mean effective pressure can only be obtained by the use of the indicator and from its diagram. The mean speed of piston is readily computed when, by counting the revolutions of the engine by the eye, watch in hand, or by the use of any convenient and reliable form of counter, the average number is obtained for the unit of time. The mean effective pressure

in pounds per square inch and the velocity of the piston in feet per minute being thus ascertained, the product of these factors into the *net* area of piston in square inches—the area of the rod being deducted on that side—gives the work, $2p_m LAN$, in foot-pounds per minute; and the indicated horse-power,

$$\text{I. H. P.} = \frac{2p_m LAN}{33000},$$

is at once given.

If working up many diagrams from the same engine, the first step should be the computation of the “constant of the engine,” a figure which expresses the horse-power of the engine, under its regular conditions of operation, for each pound mean pressure on the piston. Thus,

$$\text{I. H. P.} = \frac{2p_m LAN}{33000} = \frac{2p_m VA}{33000},$$

and when $p_m = 1$, this becomes

$$C = \frac{2LAN}{30000} = \frac{2VA}{33000},$$

where the symbols are those customarily used. Each diagram is then measured up and its mean pressure obtained, and the multiplication of this quantity by the constant thus computed gives the horse-power for that diagram.

69. The Steam or Water Consumption of the engine cannot be exactly ascertained by the use of the indicator, since a portion of the steam entering the engine is always instantly condensed by contact with the cooler walls of the cylinder, and another portion, sometimes considerable in amount, may escape past the piston, or through the valves, by leakage. The indicator does exhibit, however, the pressure and volume of steam actually present at each instant in the steam-cylinder, and it thus becomes easy to compute its weight and to obtain a measure of the quantity thus shown by the indicator, for comparison with the total quantity supplied by the boiler, and

thus to ascertain the losses, by condensation and leakage, of power, heat, and steam. The pressure being shown on the diagram at every point in the stroke, the "steam-tables" give the corresponding specific weights, the weights per unit of volume; and the space traversed by the piston up to the given point, plus the clearance-space, measures the actual volume; the latter quantity, multiplied by the specific weight, is the weight of uncondensed steam present in the cylinder at the specified point. The mean weight per stroke, multiplied by the number of strokes, being compared with the total weight supplied by the boiler in the same time, as shown by the log of the boiler-trial, the difference is the waste by internal condensation and leakage.

The real measure of the efficiency of any engine is the quantity of steam used by it to develop unity of power, and that efficiency is the greater the smaller its legitimate demand for steam, and the less its waste in these directions. Should it be impracticable to conduct a boiler-trial to determine the amount of steam drawn off to supply the engine, it may be possible to secure a fairly approximate measure—when it is known that the boiler gives dry steam—by observing the rate of fall of the water-level with the feed shut off and computing the volume and weight evaporated from the known dimensions of the volume thus traversed by the subsiding water-surface. Care must be taken not to allow its fall to go so far as to become a source of risk. It is usually easy to measure the volume corresponding to a fall nearly equal to the length of the gauge-glass. In many cases the quantity of steam shown by the indicator, at the point of cut-off, may be determined from the diagram, and a known or probably fair allowance may be added for unindicated wastes, to obtain an approximate measure of the quantity of water demanded per horse-power and per hour. This waste has been seen to be rarely as little as ten per cent., and often as much as thirty per cent. and upward.

The volume added by the clearance and port passages varies greatly with the type and build of the engine. In the single-valve and in the older forms of poppet-valve engines it

is rarely less than six, and is often ten, per cent. and more; in the best of modern engines it is often as low as two per cent. It may be easily measured, either from the drawings of the engine or by filling these spaces with water from a quantity previously weighed. The weight required to fill the clearances and ports gives the means of computing their volume from the known density of the liquid. Where, as is usual in recent and well-designed engines, considerable compression is employed, the saving of steam thus effected is to be carefully allowed for in all determinations of steam accounted for by the indicator. The loss by leakage should be inappreciable in any good engine, and this being ascertained by test, giving steam at one end and observing the escape of steam, if any occurs, by opening the indicator-cock on the other end, the whole waste shown will be due to cylinder condensation, the amount of which, as a percentage of the steam accounted for by the indicator, and as the quantity to be added to the latter for engines of fair size and good construction, may be taken as, approximately, from about $0.15 \sqrt{r}$ in the best cases of compound engines, to $0.2 \sqrt{r}$ in ordinary good unjacketed engines, and above the latter figure for engines of older type and slower speeds of rotation; r being the ratio of expansion for a single cylinder only, in the case of the compound engine, and that cylinder being taken which has the highest value of r .

When the problem to be solved is, as usual, the determination of the efficiency of any actual engine, as distinguished from the simple thermodynamic efficiency of the ideal engine, the indicator aids the engineer in its solution by showing the precise quantity of steam present at every instant during the stroke, and, hence, the quantity of water present at the same time; the sum of these two weights being always, if the piston and valves are tight, equal to the weight of feed-water passing through the boiler and entering the engine as a mixed working fluid. The volume and pressure of the steam are shown by the indicator, and the weight is easily computed from its known density at the given pressure. The portion of stroke traversed at any instant, added to the clearance-space,

measured in equivalent cylinder length, gives the volume of steam present. The quantity of steam supplied is equal to the measured quantity at the point of cut-off, less that retained by compression. The difference between the weight of steam thus measured at any point in the stroke, and the total weight of feed-water supplied, or steam entering the engine, per stroke, is the weight of water present.

Also, the total weight of mixed steam and water present from the point of cut-off to the opening of the exhaust-port is the sum of the quantity coming from the boiler and that compressed into the clearance-spaces. The variation of this quantity is well shown by the following data obtained by Mr. Spangler:*

| | | | | | |
|--------------------------|--------------|-------------------------|-------|------|-------|
| Weight of steam | per I. H. P. | per hour, | . . . | lbs. | 28.15 |
| " " priming | " " | 9 per cent., | . . . | " | 2.78 |
| " " feed-water | " " | | . . . | " | 30.93 |
| " " steam at 0.9 stroke, | | | . . . | " | 20.06 |
| " " water | " " | 33 per cent., nearly, | | " | 10.87 |
| " " steam | " 0.7 | " | . . . | " | 19.23 |
| " " water | " " | " 38 per cent., nearly, | | " | 11.70 |
| " " steam | " 0.5 | " | . . . | " | 18.27 |
| " " water | " " | " 40 per cent., nearly, | | " | 12.66 |
| " " steam | " 0.3 | " | . . . | " | 17.31 |
| " " water | " " | " 58 per cent., nearly, | | " | 13.62 |

These figures, as will be seen by comparison with other similar data, are indicative of a greater waste than usually occurs in large engines, due to the small size of that here referred to. It is evident that all variations from the proportions of the mixture entering the engine must be due to the transfer of heat to and from the metal of the cylinder and piston. The above figures show a gradual increase of the proportion of steam present produced by re-evaporation of that condensed initially, on entrance, from the point of cut-off up to the end of the stroke.

The weight of steam in the cylinder at any point of the

* Journal Franklin Institute, Feb. 1886.

stroke, in pounds per indicated horse-power per hour, is always equal to

$$\frac{W}{\text{I. H. P.}} = \frac{60 \times 2lanw'}{144} \div \frac{p_m lan}{33000} = \frac{13750w}{p_m} = w,$$

or

$$w = \frac{a}{p_m} \left[\left(\frac{1}{r} - \frac{c}{v_2} \right) w' - \frac{(c' - c)w''}{v_2} \right];$$

in which

p_m = mean effective pressure ;

c and c' = the volume of clearance-space and of steam at the point of closing of the exhaust-valve and beginning of compression ;

$\frac{c}{v_2}$ and $\frac{c'}{v_2}$ = their ratios to total cylinder volume ;

r = ratio of the given travel to full stroke of piston ;

$w, w',$ and w'' = weight of steam per indicated horse-power per hour, the specific weight at the pressure found at the given point, and the specific weight at the beginning of compression ;

a = when the piston area is measured, as above, in square inches and the stroke in feet,

$$\frac{1980000}{144} = 13750.$$

This computation is commonly made for the points of cut-off and of release. At the former the "initial condensation" is obtained, and probably the best measure of the waste by condensation ; at the latter a measure is secured of the state of the mixture exhausted from the engine.

The following example, from a diagram taken by Barrus,* employing Clarke's tables,† illustrates these computations :

Taking $\frac{1}{r} = 0.308$ at cut-off, and $\frac{1}{r} = 0.901$ at release, $c = 0.02v_2$;

* On the Tabor Indicator, p. 48.

† Manual for Mechanical Engineers.

$c' = 0.071v_2$; $p_m = 38.45$; $w' = 0.1844$ at cut-off and 0.0705 at release; $w'' = 0.0457$. Then, at cut-off,

$$w = (13750 \div 38.45)[(0.308 + 0.02)0.1844 - (0.071 + 0.02)0.0457] = 20.13 \text{ lbs. ;}$$

and, at release,

$$w = (13750 \div 38.45)[(0.91 + 0.02)0.0705 - (0.071 + 0.02)0.0457] = 21.95 \text{ lbs.}$$

Here, the indicator, in the instance from which these figures are derived, shows a difference in computed weights of steam per horse-power and per hour at the points of cut-off and release amounting to nearly two pounds—about ten per cent., which difference is the measure of the re-evaporation taking place during expansion. To the figures above obtained must be added the allowances for total wastes.

We have in many instances so little compression that it may be neglected. In such cases the following very simple process may be adopted: Assuming the working fluid to be water instead of steam, the quantity demanded would be, per horse-power per hour,

$$w_a = \frac{1980000 \times 62.4}{144} = 857900$$

pounds at one pound pressure per square inch; and, at any other pressure, p ,

$$w_b = \frac{857900}{p};$$

while if steam be employed the weight would be less in proportion to its greater, its specific, volume, v' , and the weight actually needed would be

$$w_c = \frac{857900}{p_m v'}, \text{ nearly ;}$$

which may be corrected for clearance and compression.

The detailed method of computation from this indicator-diagram is illustrated fully, later (§ 71).

The following is a convenient form of this expression for steam and water consumption:

Let p_1 = initial pressure, absolute;

p_2 = back-pressure, absolute;

r = true expansion-ratio;

c = clearance fraction;

D = density of steam in lbs. per cu. ft.;

w = weight of steam per horse-power per hour;

$$w = \frac{a}{p_1} \cdot \frac{1 + c\left(1 + \frac{p_2 r}{p_1}\right)}{(1 + \log_e r)(1 + c) - \left(\frac{p_2}{p_1} + c\right)r}.$$

The constant has been seen to have the value, in British measures, $a = 13750$. Compression is here neglected. This expression assumes a minimum value, for the ideal case, as is elsewhere seen, where

$$r = \frac{1 + c}{\frac{p_2}{p_1} + c}, \text{ nearly.}$$

This value is, in the real engine, found to be greatly reduced by the occurrence of internal wastes, by cylinder condensation or leakage.

The following table, prepared by Mr. Thompson, gives the factors employed in computing the indicated consumption of water.* The method is illustrated in Fig. 97. The mean effective pressure must be known, but the horse-power or the size of the cylinder need not be known. Draw a vertical at each end of the diagram, and continue the expansion-curve to t . From t draw tC . Measure the absolute pressure at t , and find in the table, page 244, the corresponding number. Numbers

* Hemenway's Indicator Practice; N. Y., 1886; J. Wiley & Sons. American Machinist.

WATER-COMPUTATION TABLE.

| T. P. | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-------|----------|----------|----------|----------|----------|----------|----------|----------|----------|----------|
| 3 | 117.300 | 121.015 | 124.717 | 128.406 | 132.083 | 135.748 | 139.399 | 143.075 | 146.665 | 150.279 |
| 4 | 153.880 | 157.514 | 161.137 | 164.750 | 168.353 | 171.945 | 175.527 | 179.098 | 182.659 | 186.210 |
| 5 | 189.750 | 193.336 | 196.914 | 200.483 | 204.044 | 207.598 | 211.142 | 214.679 | 218.208 | 221.728 |
| 6 | 225.240 | 228.799 | 232.351 | 235.897 | 239.437 | 242.970 | 246.497 | 250.017 | 253.531 | 257.039 |
| 7 | 260.540 | 264.056 | 267.566 | 271.071 | 274.570 | 278.063 | 281.550 | 285.031 | 288.506 | 291.976 |
| 8 | 295.440 | 298.922 | 302.400 | 305.872 | 309.338 | 312.800 | 316.256 | 319.708 | 323.154 | 326.594 |
| 9 | 330.030 | 333.488 | 336.941 | 340.389 | 343.833 | 347.273 | 350.707 | 354.137 | 357.563 | 360.984 |
| 10 | 364.400 | 367.842 | 371.280 | 374.714 | 378.144 | 381.570 | 384.992 | 388.410 | 391.824 | 395.234 |
| 11 | 398.640 | 402.064 | 405.485 | 408.902 | 412.315 | 415.725 | 419.131 | 422.534 | 425.933 | 429.328 |
| 12 | 432.720 | 436.120 | 439.517 | 442.911 | 446.301 | 449.688 | 453.071 | 456.451 | 459.828 | 463.200 |
| 13 | 466.570 | 469.950 | 473.326 | 476.699 | 480.068 | 483.435 | 486.798 | 490.159 | 493.516 | 496.869 |
| 14 | 500.220 | 503.596 | 506.968 | 510.338 | 513.706 | 517.070 | 520.432 | 523.790 | 527.146 | 530.500 |
| 15 | 533.850 | 537.213 | 540.573 | 543.930 | 547.285 | 550.638 | 553.987 | 557.334 | 560.679 | 564.011 |
| 16 | 567.360 | 570.713 | 574.063 | 577.411 | 580.757 | 584.100 | 587.441 | 590.780 | 594.115 | 597.449 |
| 17 | 600.780 | 604.109 | 607.435 | 610.759 | 614.081 | 617.400 | 620.717 | 624.031 | 627.343 | 630.653 |
| 18 | 633.960 | 637.265 | 640.567 | 643.867 | 647.165 | 650.460 | 653.753 | 657.043 | 660.331 | 663.617 |
| 19 | 666.900 | 670.200 | 673.498 | 676.793 | 680.086 | 683.378 | 686.666 | 689.953 | 693.238 | 696.520 |
| 20 | 699.800 | 703.098 | 706.394 | 709.688 | 712.980 | 716.270 | 719.558 | 722.844 | 726.128 | 729.410 |
| 21 | 732.690 | 735.968 | 739.244 | 742.518 | 745.790 | 749.060 | 752.328 | 755.594 | 758.858 | 762.120 |
| 22 | 765.380 | 768.660 | 771.938 | 775.215 | 778.490 | 781.763 | 785.034 | 788.303 | 791.570 | 794.836 |
| 23 | 798.100 | 801.362 | 804.622 | 807.881 | 811.138 | 814.393 | 817.646 | 820.897 | 824.146 | 827.394 |
| 24 | 830.640 | 833.908 | 837.175 | 840.440 | 843.703 | 846.965 | 850.225 | 853.484 | 856.741 | 859.996 |
| 25 | 863.250 | 866.502 | 869.753 | 873.002 | 876.249 | 879.495 | 882.739 | 885.982 | 889.223 | 892.462 |
| 26 | 895.700 | 898.936 | 902.171 | 905.404 | 908.635 | 911.865 | 915.093 | 918.320 | 921.545 | 924.768 |
| 27 | 927.990 | 931.210 | 934.429 | 937.646 | 940.861 | 944.075 | 947.287 | 950.498 | 953.707 | 956.914 |
| 28 | 960.120 | 963.352 | 966.583 | 969.813 | 973.041 | 976.268 | 979.493 | 982.717 | 985.939 | 989.160 |
| 29 | 992.380 | 995.598 | 998.815 | 1002.031 | 1005.245 | 1008.458 | 1011.669 | 1014.879 | 1018.087 | 1021.294 |
| 30 | 1024.500 | 1027.704 | 1030.907 | 1034.109 | 1037.309 | 1040.508 | 1043.705 | 1046.901 | 1050.095 | 1053.288 |
| 31 | 1056.480 | 1059.670 | 1062.859 | 1066.047 | 1069.233 | 1072.418 | 1075.601 | 1078.783 | 1081.963 | 1085.142 |

WATER-COMPUTATION TABLE—Continued.

| T. P. | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-------|----------|----------|----------|----------|----------|----------|----------|----------|----------|----------|
| 32 | 1088.320 | 1091.528 | 1094.736 | 1097.942 | 1101.146 | 1104.350 | 1107.552 | 1110.754 | 1113.954 | 1117.152 |
| 33 | 1120.350 | 1123.546 | 1126.742 | 1129.936 | 1133.128 | 1136.420 | 1139.510 | 1142.700 | 1145.888 | 1149.074 |
| 34 | 1152.260 | 1155.444 | 1158.628 | 1161.810 | 1164.990 | 1168.170 | 1171.348 | 1174.526 | 1177.702 | 1180.876 |
| 35 | 1184.050 | 1187.222 | 1190.394 | 1193.564 | 1196.732 | 1199.900 | 1203.066 | 1206.232 | 1209.396 | 1212.558 |
| 36 | 1215.720 | 1218.917 | 1222.112 | 1225.307 | 1228.500 | 1231.693 | 1234.884 | 1238.075 | 1241.264 | 1244.453 |
| 37 | 1247.640 | 1250.827 | 1254.012 | 1257.197 | 1260.380 | 1263.563 | 1266.744 | 1269.925 | 1273.104 | 1276.283 |
| 38 | 1279.460 | 1282.637 | 1285.812 | 1288.987 | 1292.160 | 1295.333 | 1298.504 | 1301.675 | 1304.844 | 1308.013 |
| 39 | 1311.180 | 1314.347 | 1317.512 | 1320.677 | 1323.840 | 1327.003 | 1330.164 | 1333.325 | 1336.484 | 1339.643 |
| 40 | 1342.800 | 1345.957 | 1349.112 | 1352.267 | 1355.420 | 1358.573 | 1361.724 | 1364.875 | 1368.024 | 1371.173 |
| 41 | 1374.320 | 1377.467 | 1380.612 | 1383.757 | 1386.900 | 1390.043 | 1393.184 | 1396.325 | 1399.464 | 1402.603 |
| 42 | 1405.740 | 1408.877 | 1412.012 | 1415.147 | 1418.280 | 1421.413 | 1424.544 | 1427.675 | 1430.804 | 1433.933 |
| 43 | 1437.060 | 1440.230 | 1443.398 | 1446.566 | 1449.734 | 1452.900 | 1456.066 | 1459.230 | 1462.394 | 1465.558 |
| 44 | 1468.720 | 1471.882 | 1475.042 | 1478.202 | 1481.362 | 1484.520 | 1487.678 | 1490.834 | 1493.990 | 1497.146 |
| 45 | 1500.300 | 1503.454 | 1506.606 | 1509.758 | 1512.910 | 1516.060 | 1519.210 | 1522.359 | 1525.506 | 1528.654 |
| 46 | 1531.800 | 1534.946 | 1538.090 | 1541.234 | 1544.378 | 1547.520 | 1550.662 | 1553.802 | 1556.942 | 1560.082 |
| 47 | 1563.220 | 1566.358 | 1569.494 | 1572.630 | 1575.766 | 1578.900 | 1582.034 | 1585.166 | 1588.298 | 1591.430 |
| 48 | 1594.540 | 1597.690 | 1600.818 | 1603.946 | 1607.074 | 1610.200 | 1613.326 | 1616.450 | 1619.574 | 1622.698 |
| 49 | 1625.820 | 1628.942 | 1632.062 | 1635.182 | 1638.302 | 1641.420 | 1644.538 | 1647.654 | 1650.770 | 1653.886 |
| 50 | 1657.000 | 1660.114 | 1663.226 | 1666.338 | 1669.450 | 1672.560 | 1675.670 | 1678.778 | 1681.886 | 1684.994 |
| 51 | 1688.100 | 1691.206 | 1694.310 | 1697.414 | 1700.518 | 1703.620 | 1706.722 | 1709.822 | 1712.922 | 1716.022 |
| 52 | 1719.120 | 1722.218 | 1725.314 | 1728.410 | 1731.506 | 1734.600 | 1737.694 | 1740.786 | 1743.878 | 1746.970 |
| 53 | 1750.060 | 1753.150 | 1756.238 | 1759.327 | 1762.414 | 1765.500 | 1768.586 | 1771.670 | 1774.754 | 1777.838 |
| 54 | 1780.920 | 1784.002 | 1787.082 | 1790.162 | 1793.242 | 1796.320 | 1799.398 | 1802.474 | 1805.550 | 1808.626 |
| 55 | 1811.700 | 1814.829 | 1817.957 | 1821.084 | 1824.211 | 1827.338 | 1830.463 | 1833.588 | 1836.713 | 1839.837 |
| 56 | 1842.960 | 1846.083 | 1849.205 | 1852.326 | 1855.447 | 1858.568 | 1861.687 | 1864.806 | 1867.925 | 1871.043 |
| 57 | 1874.160 | 1877.277 | 1880.393 | 1883.508 | 1886.623 | 1889.738 | 1892.851 | 1895.964 | 1899.077 | 1902.189 |
| 58 | 1905.300 | 1908.411 | 1911.521 | 1914.630 | 1917.739 | 1920.848 | 1923.955 | 1927.062 | 1930.169 | 1933.275 |
| 59 | 1936.380 | 1939.485 | 1942.589 | 1945.692 | 1948.795 | 1951.898 | 1954.999 | 1958.100 | 1961.201 | 1964.301 |
| 60 | 1967.400 | 1970.499 | 1973.597 | 1976.694 | 1979.791 | 1982.888 | 1985.983 | 1989.078 | 1992.173 | 1995.267 |

under T. P. represent the terminal pressure in pounds, and the figures, 1, 2, 3, etc., tenths of a pound.

Divide this number by the mean effective pressure; the quotient will be the steam accounted for per horse-power per

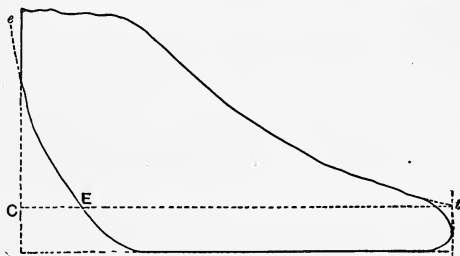


FIG 97.—INDICATED STEAM VOLUMES. SCALE 40.

hour, uncorrected for compression. To make this correction multiply by tE and divide by tC .

When the maximum compression is not as high as the terminal pressure, the compression-curve must be extended, as e , and E will be outside the diagram.

In illustration of the computation of the economical performance of engines by determining their expenditure of heat, measured in British thermal units, may be given the following, as computed by Mr. Barrus,* using Clark's tables.†

Assume a non-condensing engine to use 27 lbs. of feed-water per horse-power per hour, supplied at 212° F.; a condensing engine 18 lbs., at 130° ; and a compound engine 14 lbs., at 170° . The pressure in the first two cases is 80 lbs. and in the last case 120. In the first two cases the steam contains $\frac{1}{2}$ per cent moisture, and in the last case it is superheated 20° . The total heat of saturated steam of 80 lbs. pressure [94.7 lbs. absolute] is 1212.2 *B. T. U.* Deduct the heat corresponding to 0.05 moisture, $0.05 \times 885.9 = 4.4$ [885.9 being the latent heat], and there remain 1207.8 units, the total heat of steam

* The Tabor Indicator.

† Manual for Mechanical Engineers.

containing $\frac{1}{2}$ of one per cent of moisture, measured above zero. Deduct the heat corresponding to a feed-water temperature of 212° , 212.9 thermal units, and there remain 994.9 units, the total heat of one pound of steam, containing $\frac{1}{2}$ of one per cent moisture, above the temperature of feed-water. Multiply this by 27, and the product, 26,862.3 units, is the heat expended per horse-power per hour.

A similar computation gives 19,400.4 thermal units per horse-power per hour for the second case.

In the third case, the total heat of saturated steam of 120 lbs. pressure [134.7 lbs. above zero] is 1220.1 *B. T. U.* The heat corresponding to 20° superheating is $20 \times 0.475 = 9.5$, which gives 1229.6 units for the total heat of superheated steam. Deduct 170.4 thermal units, the heat corresponding to a feed-water temperature of 170° , and multiply by 14, and we have for a product 14,828.8 units, the heat expended per horse-power per hour.

These results are tabulated below :

| KIND OF ENGINE. | Non- Condensing. | Condensing. | Compound. |
|----------------------------------------------------------------|---------------------|-------------|-------------|
| | A. | B. | C. |
| Boiler-pressurelbs. | 80 | 80 | 120 |
| Av. temp. of feed-water.deg. | 212 | 130 | 170 |
| Feed-water per H. P. per hour.....lbs. | 27 | 18 | 14 |
| Percentage moisture in steam..... % | 0.5 | 0.5 | 20° superh. |
| Total heat of saturated steam.....th. un. | 1212.2 | 1212.2 | 1220.1 |
| Total heat corrected for moisture and superheating.....th. un. | 1207.8 | 1207.8 | 1229.6 |
| Heat of feed-water... ..“ | 212.9 | 130 | 170.4 |
| Heat expended per pound.....“ | 994.9 | 1077.8 | 1059.2 |
| Heat expended per H. P. per hour....“ | 26,862.3 | 19,400.4 | 14,828.8 |

A comparison of the heat thus computed, as expended, with the heat-equivalent of the useful work performed, determines the efficiency.

As each horse-power is the thermal equivalent of 42.75 heat-units per minute or 2565 units per hour, we have for the three cases,

$$E: \quad \frac{A.}{2565} = \frac{2565}{26862} = 0.096; \quad \frac{B.}{2565} = \frac{2565}{19400} = 0.137; \quad \frac{C.}{2565} = \frac{2565}{14829} = 0.16;$$

or efficiencies of 9.6, 13.7, and 16 per cent., as compared with an engine of efficiency unity, perfectly utilizing all the heat-energy supplied to it. This is the method first adopted by Rankine, except that thermal, rather than mechanical, units are employed.

70. Constructing Hyperbolic Curves, such as are commonly taken to represent the variations of pressure and volume in the ideal diagram, enables the engineer to obtain some idea of the method and extent of variation of the actual quantities in real engines from those of the ideal case. There are several methods of constructing these curves, of which the simplest are, perhaps, the following, as applied to produce the equilateral hyperbola, the curve of Mariotte, to which the expansion-line, in the best classes of engine, very closely approximates, and which is commonly taken as the standard.

Let XX , YY be given asymptotes (i.e., the clearance and true vacuum lines of the indicator-card) and x any given point, and let xx , xy be its co-ordinates.

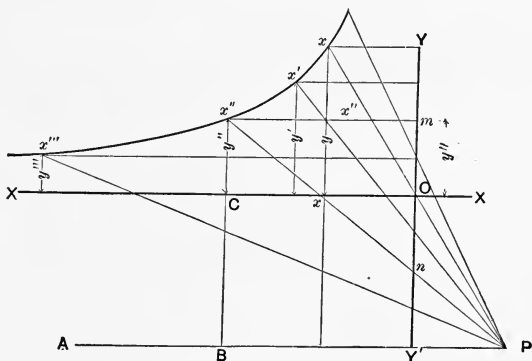


FIG. 98.—THE HYPERBOLA.

Extend YO until $OY' = YO$ and draw AP , making $Y'P$ equal to xY and parallel to XX .

Divide YO and OY' into similar divisions.

Assume an ordinate Om of a point to be found, and draw mx'' parallel to XX .

At Y' erect $Y'n = Om$, and draw Pnx'' ; the point x'' of intersection with $x''n$ is the required point.

For in the triangles $ny'P$, $nm x''$ we shall have

$$nY' : Y'P :: mn : x''m = \frac{xy}{y''} = x'';$$

$$\text{i.e., } y'' : x :: y : x''. \quad \text{Q. E. D.}$$

When the expansion-line is true to the hyperbolic curve, it becomes possible to obtain a fairly approximate measure from the diagram of the clearance-space; or, the latter being known, to determine the real locus of the hyperbolic expansion-curve, as follows:

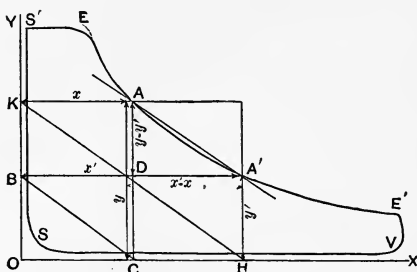


FIG. 99.—THE HYPERBOLIC EXPANSION-LINE.

Let S', E, E', V, S represent an indicator-card; let OX be the line of perfect vacuum; OY the line at end of cylinder plus the clearance; then, OX and OY will be asymptotes of the hyperbola E, A, A', E' , the curve of expansion.

Take two points on the curve AA' , and $AK, AC, A'B$, and $A'H$ will be their co-ordinates.

Draw AA' , and from C , the line CB parallel to AA' ; the point B , where it intersects $A'B$, will be a point in the line OY .

Or, draw HK parallel to AA' , and K , the intersection with AK , will be such a point.

For by Mariotte's law and from the properties of the hyperbola $xy = m$; $x'y' = m$; $\therefore xy = x'y'$.

$$\therefore x : x' :: y' : y; \quad x' - x : x :: y - y' : y';$$

$$\text{or, } A'D : BD :: AD : DC.$$

And, from similar triangles (by construction),

$$A'D : BD :: AD : DC'. \quad Q. E. D.$$

Conversely, having given the clearance and the scale of the indicator, with point of cut-off, to find the *expansion-line*.

In proportion $y - y' : y' :: x' - x : x$, assume x' and find values of y' by constructing the triangle KPH , similar to ADA' .

Taking the point of release as a point in the hyperbolic curve, and laying down that curve on the diagram, it will be found, not only that the curve and the expansion-line of the diagram do not coincide, but that the latter falls above the former throughout its length, in nearly all cases, indicating, usually, initial condensation and later re-evaporation, but sometimes indicating some leakage as well. If the weight of steam actually drawn from the boiler be taken as the basis of a diagram, using its volume as the initial ordinate of the hyperbolic curve, it becomes easy to trace the variations of the whole actual diagram from the ideal indicator-card, as here shown.

In any case in which the curve represented by the expansion-line is of the class of which the equation is

$$pv^n = p_1v_1^n = p_2v_2^n,$$

the co-ordinates sought, any one point, p_1v_1 or p_2v_2 being given, may be found, and any new point in the ideal curve determined by computation, thus: From the above expression,

$$n \log v + \log p = n \log v_1 + \log p_1;$$

and if p_1 and v_1 are known, for any assumed volume v , the logarithm of the corresponding new pressure must be

$$\log p = n \log v_1 + \log p_1 - n \log v;$$

which expression being used to determine several points, the curve may be drawn through them.

The values of n have been seen to be as follow:

| | |
|-------------------------------------------------|--------------|
| Equilateral hyperbola, | 1 |
| Curve of steam; saturation $\frac{17}{16}$, or | 1.0646 |
| Adiabatic curve, steam, | 1.035 + 0.1x |
| “ “ gas, | 1.408 |
| Isothermal “ “ | 1.0 |

The variation of the actual ratios of expansion from their apparent values, in engines having large clearance-spaces, is very considerable at high ratios of expansion and in short-stroke engines. The following table (p. 252), published by Mr. Grimshaw, is sufficiently extensive for ordinary purposes, and well exhibits those differences.*

The close approximation of the three principal steam-ex-

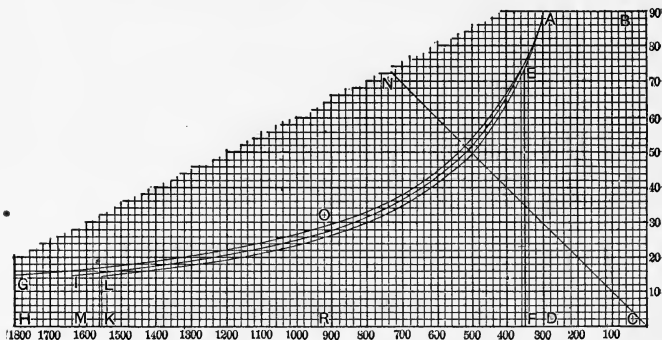


FIG. 100.—THE THREE EXPANSION-CURVES.

pansion lines is well shown by the accompanying diagram, a set of curves shown in various publications, but probably first laid down in this form by Mr. Porter.† AB exhibits the initial volume, as does also CD ; AD and BC represent the initial pressure; EF is an ordinate, taken at convenience; and the terminal ordinates are GH , IM , and LK . OR is taken at half-stroke; while CN is the axis of the equilateral hyper-

* Am. Machinist, Jan. 20, 1883, p. 5.

† Steam-engine Indicator, p. 123.

bola, AOG , the upper curve, of which CB and CH are asymptotes. Ordinates measure absolute pressures in pounds per square inch; abscissas represent volumes of unity of weight (1 lb.). Thus BA is the volume (4.73 cu. ft.) of one pound of steam at a total pressure of 90 pounds per square inch; $ABCD$ is the external work done in its production. It is this curve which is commonly assumed to be that of the expansion of steam.

The curve AOI is the curve of dry and saturated steam, its co-ordinates representing the simultaneous pressure and volume of the fluid when in contact with the mass of water from which it is produced. The expansion is less, and the rate of fall of pressure greater, than if it were to follow the law of Mariotte. It is this curve which is assumed to be described when steam expands in well-jacketed engines.

The lower line, AOL , is the adiabatic curve, assumed to be obtainable in engines with non-conducting cylinders and approximately in "high-speed engines." The area under this, as under the other curves, represents the work done as the steam expands, and exhibits the gain obtainable by expansion, in each case. In all real engines, however, the expansion-line falls at first more rapidly, and finally more slowly, than either of these curves. As elsewhere seen, this variation from the ideal curve is often very observable.

71. Cylinder Condensation and Leakage produce variations in the diagram, as obtained, which differently affect the different parts of the curve. Leakage can usually be eliminated, and always should be before the engine is set at work regularly. The first-named waste is usually irremediable. When the exact measure of the quantity of steam expended is obtained by a boiler-trial, it is easy to trace these variations, as in the diagram here given, as taken from the engine and worked up by the late Professor C. A. Smith, in which illustration the diagram which should have been produced by the same steam, had there been no initial condensation, is shown with the real diagram.*

* Steam-making, p. 91.

This indicator-diagram is an unusually good sample, as to form, and was taken from the St. Louis high-service pumping-engine, a machine of 705 I. H. P., 85 inches diameter of cylinder, and 10 feet stroke of piston, making $11\frac{1}{2}$ revolutions per minute. Taking measures of the abscissas of the two diagrams, it is seen that the condensation amounts to from about 30 per cent. as a minimum to 50 per cent. as a maximum, so far as measurable, the actual card illustrating the expansion in a metallic cylinder of the steam, which would have given the larger diagram in an ideal engine with its non-conducting cylinder. The complete ideal diagram would extend proportionally farther toward the right and beyond the limits of the actual figure. When the two lines continue so far separated,

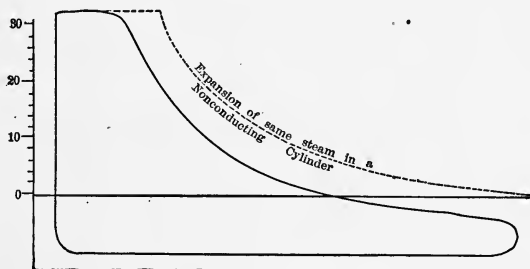


FIG. 101.—THE REAL AND THE IDEAL CARD.

it is an indication of large initial condensation, and correspondingly great re-evaporation after the exhaust-valve opens; as the initial condensation is due to, and is proportional to, the re-evaporation. In most cases, however, the engineer, unable to determine these data, assumes the point of release, or the point of intersection of the expansion-line prolonged with the ordinate at the extreme end of the diagram, as that of coincidence of the ideal and the real curve, and draws the hyperbolic curve backward from that as a given point, in the manner already described. A comparison of the ideal diagram thus formed with the actual indicator-card will give a means of judging of the character of the engine studied as a thermo-

dynamic apparatus, and of comparing different engines. An exact coincidence of the two diagrams, in any given case, would not prove, or even give a presumption of freedom from such waste; nor would the equality, in this respect, of diagrams from any two engines prove more than a probable general similarity in their performance, thermodynamically. Such comparisons are, nevertheless, both interesting and instructive, as is seen in the following examples. They also give some indications of the probable consumption of water and steam, the real gauge of the efficiency of engines. The clearance may be determined by measurement of the engine, by the graphical method described in the preceding article, or by the following simple methods of construction.*

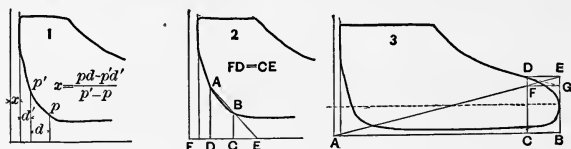


FIG. 102.—IDEAL CONSTRUCTIONS.

In case 1 let p and d , p' and d' , be co-ordinates of the two given points, and x = the clearance; then

$$(x + d)p = (x + d')p', \quad \text{and} \quad x = \frac{pd - p'd'}{p' - p}.$$

Or we may determine the clearance geometrically by the following construction (see case 2). Assume two points A and B in the compression-curve; connect them by a right line, AB , continuing this line until it cuts FE at E . Draw AD and BC perpendicular to FE , and make $FD = CE$. Then F is the end of the ideal diagram including clearance, and the distance of F from the end of the indicator-diagram is the clearance.

To lay out the theoretical diagram: Draw a line representing the boiler-pressure and also a line of perfect vacuum, at 14.7 pounds below the atmospheric line, unless the true baro-

* First published by Mr. G. H. Babcock; Journal Franklin Institute, Sept. 1869.

metric reading is given. Next divide the length of the diagram, including clearance, into any number of equal parts, as ten. Measure the pressure at the point of release, and find the terminal pressure by any convenient method as that shown in case 3, in which *AB* is the length of the diagram, including clearance, and *D* is the point of release. Draw *DE* parallel to *AB* and join *AE*, cutting *DC* at *F*. Draw *FG* parallel to *AB*, and *BG* will represent the terminal pressure, the tension at which a quantity of steam equal to the whole capacity of the

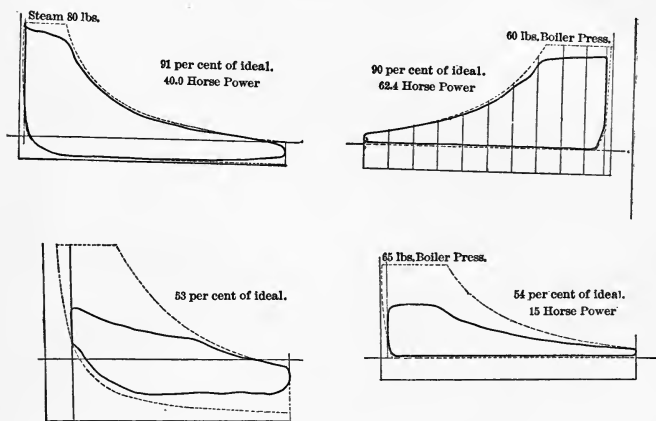


FIG. 103.—IDEAL AND REAL DIAGRAMS.

cylinder and clearance would be discharged at the termination of the stroke.

The pressure at any other point of the stroke is easily found by the usual methods. With ten divisions, the several ordinates of the expansion-curve may be obtained by multiplying the terminal pressure by the following factors: 1, 1.11, 1.25, 1.429, 1.667, 2, 2.5, 3.333, 5, 10. Having found the ideal pressure at each division, we trace a curve through these points and determine the ideal point of cut-off, giving the same terminal pressure as is observed in the actual case.

If the exhaust-valve closes before the end of the return stroke, so much of the cylinder full of steam as is thus imprisoned must be allowed for in the ideal diagram. Draw a hyperbolic curve tangent to the actual compression-line, and extending to the line of boiler-pressure, and thus find the boundary of the ideal diagram.

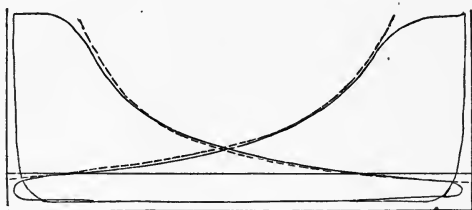


FIG. 104.—IDEAL AND REAL CARDS.

The group of four diagrams, Fig. 103, is given by Mr. Babcock in illustration of this method. The upper pair show a remarkable approximation of the actual to the standard figure, each giving, from the measured steam, 90 per cent. of the power which an engine having a non-conducting cylinder should give. One is a condensing, the other a non-condensing, mill engine; both designed by Mr. Babcock. The other pair are similar in their wastefulness, each giving but about one half the maximum, ideal, amount of work. One is from an old naval condensing engine, the other from a non-condensing stationary engine.

The next figure is a *fac-simile* of a pair of diagrams from an engine designed by Mr. J. W. Thompson, as studied by Mr. Hill, who gives the following analysis, using the curve of dry and saturated steam, having the equation $pv^{1\frac{1}{2}} = \text{constant}$ as the standard. The engine was 22 inches diameter of cylinder, and 44 inches stroke of piston, making 70 revolutions per minute. The clearance is stated at .0175 piston-displacement.

The diagrams were measured with an "Amsler planimeter," and read as follows:

Mean effective pressure above atmosphere (both diagrams).....19.9765 lbs.

Mean effective pressure below atmosphere (both diagrams).....10.143 lbs.

Together.....30.119 lbs. per sq. inch.
Power independent of vacuum, "constant" $\times p_m$,

$$5.9101 \times 19.9765 = 118.063 \text{ H. P.}$$

Power due vacuum,

$$5.9101 \times 10.143 = 59.946 \text{ H. P.}$$

Combined power, 178.009 H. P.

Ratio of power below atmosphere to power above atmosphere,

$$\frac{59.946 \times 100}{118.063} = 50.774 \text{ per centum.}$$

The total diagram including cushion reads 31.134 lbs., and the efficiency of cylinder becomes

$$\frac{31.134 - 30.119}{31.134} = .0326;$$

and $1 - .0326 \times 100 = 96.74$ per centum of total capacity utilized.

The expenditure of steam to produce the power according to the diagram is estimated as follows:

$$\frac{380.13 \times 44 \times 70 \times 2 \times 60}{144 \times 12} = 81305.532 \text{ cu. ft.}$$

total piston-displacement per hour.

The release by the diagrams (both ends of cylinder) appears to occur at 43.175 inches from beginning of stroke, hence

$$\frac{81305.532 \times 43.175}{44} = 79781.05 \text{ cu. ft. to release.}$$

The exhaust closes (both ends of the cylinder) at 4.1712 inches from end of stroke (return); hence

$$\frac{81305.532 \times 4.1712}{44} = 7707.764 \text{ cu. ft.}$$

Clearance-volume $81305.532 \times .0175 = 1422.846$ cu. ft.

The volume of steam accounted for to release becomes

$$79781.05 + 1422.846 = 81203.896 \text{ cu. ft.,}$$

and the volume of steam retained in the cylinder by closure of exhaust becomes

$$7707.764 + 1422.846 = 9130.61 \text{ cu. ft.}$$

The terminal pressure is $\frac{11.5 + 12.75}{2} = 12.125$ lbs.,

and the weight of a cubic foot of steam at this pressure is obtained by Tate's formula, thus: 12.125 lbs. = f = 24.7 inches mercury; and a cubic foot of water at maximum density weighs, according to Berzelius, 62.388 lbs.; hence

$$\frac{62.388}{25.62 + \frac{49513}{f + .72}} = .0316 \text{ lbs.,}$$

and $81203.896 \times .0316 = 2566.043$ lbs. steam.

The steam retained by cushioning is as follows: The pressure in front of piston at time exhaust closes (both ends of cylinder) is 3.75 lbs., and the weight per cubic foot of steam at this pressure is

$$\frac{62.388}{25.62 + \frac{49513}{7.639 + .72}} = .01048 \text{ lbs.};$$

hence, $9130.61 \times .01048 = 95.688$ lbs. steam retained by cushioning. Net steam consumed per hour,

$$2566.043 - 95.688 = 2470.355 \text{ lbs.};$$

and steam (water) per indicated horse-power per hour by the diagrams, $\frac{2470.355}{178} = 13.878$ lbs. The effective vacuum was 20.66 inches, and the losses by leakage and extra condensation were estimated as probably 15 per centum, hence

$$\frac{13.878}{.85} = 16.327 \text{ lbs.},$$

estimating an evaporative efficiency of connected boilers of 9 to 1 of coal; the cost of coal per I. H. P. per hour becomes 1.814 lbs. This is probably too low an estimate of this waste. Taking it, however, as even double this amount, 30 per cent., the coal and water consumption would be, respectively, but 2.2 and 19.826 lbs. per I. H. P. per hour; low figures, both.

The next illustration, a diagram published by Mr. Porter, as taken from a high-service pumping-engine at Providence, R. I., when making but one revolution per minute, exhibits the enormous extent to which initial condensation and later re-evaporation can occur, most remarkably.



FIG. 105.—CONDENSATION AND RE-EVAPORATION.

The hyperbolic line is at AB , and the magnitude of the terminal ordinate of the diagram, as compared with the ordinate of the hyperbola, measures the proportion of re-evaporation. It is seen that more than three times as much steam must have been condensed at entrance as remained, to produce the diagram, this proportion, at least, being later re-evaporated.

The following are the quantities of steam found at various

parts of the stroke of a compound Corliss engine, as reported by Mr. Hoadley: *

| H. P. Cut-off. | Steam, lbs. | |
|------------------|-------------|------------|
| | Present. | Condensed. |
| 0.178 | 9.97 | 6.80 |
| .625 | 11.32 | 5.18 |
| .750 | 11.35 | 5.15 |
| 1.000 | 11.35 | 5.15 |
| L. P. Cyl., end. | 10.57 | 5.93 |

The condensation in cylinder and jackets was about one-half throughout.

The next diagram, Fig. 106, illustrates the action of the air-compressor. The isothermal lines, which are here hyperbolic, are drawn from the atmospheric line as its starting-point. Two diagrams are shown superposed—the one of a common and somewhat inefficient compressor, the other of a more per-

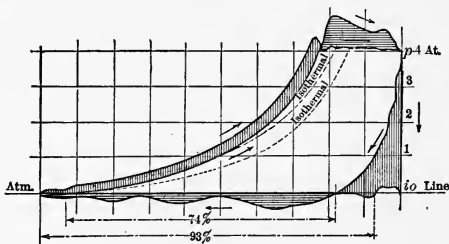


FIG. 106.—AIR-COMPRESSOR CARDS.

fect form. The former gives a diagram having an efficiency but 74 per cent. of the ideal, and the latter 93 per cent. Here the actual diagrams exceed the ideal in area, the heat of compression carrying its compression-line above the isothermal, and the defects of construction and operation of the induction and eduction valves throwing the delivery-line above the limit of pressures in the receiving reservoir.

* Steam-engine Practice in the U. S., 1884.

CHAPTER VII.

ENGINE-FRICTION ; DYNAMOMETERS.

72. Engine-friction is an important element of waste in all engines. The resistance of the engine due to internal friction and the effort demanded for its impulsion are measured, according to size, type, and condition, by from about one pound on the square inch of piston in the best large and well-designed engines, to three or four pounds, and even more, in small and inefficient machines. An efficiency of machine exceeding 90 per cent. is rare, and is considered high.

The efficiency of mechanism, the ratio of work done by the engine to the work performed by the steam on its piston, is in rare cases 95 per cent., in usual good practice about 85 or 90, and in fairly good work 85 per cent., or less. If the figure falls under 0.80, it is regarded as decidedly low.

Before and during a trial, especially, the lubrication should be seen to be thoroughly efficient. A good lubricant should be chosen, and it should be properly applied and freely used.

73. Indicated Work less Engine-friction constitutes the net useful work of the machine. The power of the engine measured, not in its working cylinder, but as delivered from its crank-shaft, is that with which its proprietor and the engineer are most concerned. A complete engine-trial, therefore, includes a careful and exact measurement of the friction of the engine, and the so-called dynamometric power of the machine, as well as the indicated power recorded in the diagram. The friction of engine is sometimes allowed for when its direct measurement is impracticable, by assuming a certain pressure as suf-

ficient to overcome its resistance, which pressure ranges from one or two pounds per square inch in large engines, to three or four in smaller sizes, the experience and judgment of the observer being taken as guides. It has been found that engine-friction may usually be taken as constant at all loads.

In his various papers on this subject, the Author first called attention to the fact that the variation of load in steam-engines is not productive either of the method or of the amount of engine-friction that has been commonly assumed by earlier authorities on that subject.* It was shown that the formula of De Pambour, which makes the internal friction of the engine proportional to the load on its piston, is not usually correct, and probably is never so, with any familiar form of engine, or under any conditions often met with in practice. It was further shown that, under the conditions of usual practice, and at all ordinary speeds and pressures of steam, the resistance of the engine itself, its internal friction, remains sensibly constant, and that the so-called friction-card of the machine represents practically the friction of the engine when fully loaded, the indicated power without load being sensibly the measure of the wasted work of the engine when in operation under load of whatever amount. Even the compound engine, contrary to the expectation of the Author, exhibited substantially the same internal friction at all loads up to its full rated power, and with no load at all. He has shown the engine-friction to be independent of the load, but to be a function of the characteristics of the engine itself, of the speed of piston and rotation, of the steam-pressure, and of the method of steam-distribution, the two last-named conditions having slight effect, the others being most important. The weight and design, and the character of the workmanship of the engine, primarily determine the amount of its internal friction; the resistance is also a direct function of its speed, and it is slightly and observably affected, within

* Friction of Non-condensing Engines. Trans. Am. S. M. E., Vol. VIII, No. CCXXVIII, and Vol. IX, No. CCLXV.

limits, by the steam-pressure variations, and by the character of valve-gear and of steam distribution and of regulation of engine. The speed and weight of the running parts of the engine may, so far as can now be ascertained, be taken as the elements controlling friction of the machine. This leads to the conclusion that the friction coefficient of the rubbing surfaces decreases with the load on the engine and with increase of pressure on them, a result confirmed by numberless experiments of the Author and others, independently. With good lubrication, the coefficient of friction rapidly decreases with intensifying pressures, and to such an extent as to make the actual resistance to movement often very nearly constant.

74. Measurements of Gross and Net Power are commonly made by means of the indicator and either the absorbing or the transmitting dynamometer, the former giving the gross or indicated power, the latter showing the amount of power applied by the engine to a brake or to its special purposes, and capable of doing useful work. It is only when such measurements of actually applied, net, power are made, that the real value of the engine as a motor can be ascertained. The efficiency of the engine as a machine, also, is thus determinable, and is measured by the ratio of the dynamometric to the indicated power; this ratio is usually about 80 per cent., but sometimes exceeds 90.

Transmitting Dynamometers are of various types and forms, but all consist of a set of pulleys so arranged that they may be placed between the prime motor and the machinery to be driven by it, while the effort is measured by, usually, a set of springs interposed between the receiving and the delivering pulleys. The magnitude of this effort is often, perhaps generally, automatically recorded on a travelling ribbon or strip of paper, and the speed of the machine is observed. The product of the effort into the velocity of the point at which it is measured is the measure of the work done in the unit of time and of the power expended. There are many forms of this instrument, but the class most generally known is probably that of

General Morin, as built in the Sibley College shops, Fig. 107, in which *A* is a pulley fast on the shaft, and *C* is a loose pulley on the same shaft, the motion being transmitted from the prime mover to one or other, according as it is desired to drive the shaft or not.

A pulley, *B*, on the same shaft, carries the belt which transmits motion to the driven machine. This pulley is loose on the shaft, so that it is capable of moving backwards and forwards through a small arc, to admit of the deflection of a spring by which the effort is transmitted from the shaft to the pulley. One end of that spring is fixed so that the blade projects like

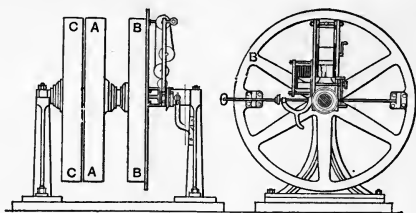


FIG. 107.—MORIN DYNAMOMETER.

an arm, and rotates with it. The other end is connected with *B*, so that the spring undergoes deflection proportional to the effort exerted by the shaft on the pulley. A frame, rotating along with it, carries an apparatus for making a band of paper move radially with a velocity proportional to the speed with which it rotates. A pencil carried by this frame traces a zero line, and another pencil carried by the spring traces a line whose ordinates represent the forces exerted. The mechanism for moving the paper is driven by a toothed ring surrounding the shaft, and kept at rest while the shaft rotates by means of a catch. When that catch is drawn back, the toothed ring rotates with the machine, and the paper is thus stopped when desired.

The Batchelder or Francis dynamometer, as designed by

Mr. Webber, is of the form shown in the following illustration. The principle of this machine was originally invented by a Mr.

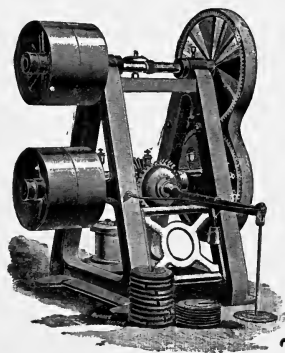


FIG. 108.—WEBBER'S DYNAMOMETER.

Samuel White, in England, in 1780-90; and was brought over to this country by Mr. Samuel Batchelder in 1836. It is said that one of these machines has been used fifteen years, weighing over one hundred and fifty thousand horse-power in small amounts, and the totals, in some cases aggregating two hundred and fifty horse-power, have substantially agreed with the results obtained from indicator-cards taken from the engine driving the same machinery.

A transmitting dynamometer, another of many forms now obtainable, is shown in the next figure. This form is often employed for light work, as in determining in detail the power consumed by each of the several machines driven by the prime motor; and this has also been used by the Author with satisfactory results. As here seen, the pulley *A* is loose on the shaft, and receives the power. Its connection with the shaft is made by means of the spider *J*, which is keyed or screwed firmly to the shaft in close contiguity with the receiving pulley, its hub, in fact, forming one of the guides to the position of the pulley on the shaft. To connect this spider with the loose receiving pulley *A*, a bell-crank lever is pivoted into projecting ears on the rim of the wheel *A*, on opposite sides, the long arm of which connects with an annular slotted collar on the shaft by means of short bars *B, B*. The short arms of the bell-crank levers connect on the inside of the fixed wheel with two radial bars, one parallel to the outer arm of the bell-crank, and the other at right angles to it, receiving near its upper end a pivot passing through a swivel hung to the arm of the spider wheel, and having its extreme end pivoted to a stud fixed on the inner side of the

rim of the receiving pulley. It will be seen that the strain of the power received through the belt on the pulley will necessarily react on the levers, and through them on the spider, which may be considered nothing more nor less than a support to these levers in sustaining them in position to connect the loose receiving pulley with the shaft. It will be seen that the levers are connected by pivots with the sliding collar *BC*, in

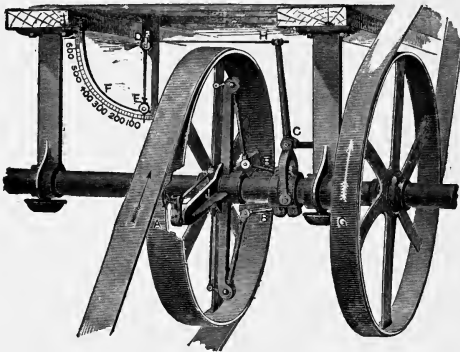


FIG. 109.—THE EMERSON DYNAMOMETER.

the annular groove of which is seated a strap, with which is connected a forked lever, *CH*. To the end of the long arm of this lever a rod, *F*, with a short section of machine-chain is attached. This chain runs over the cylindrical head *D* of a pendulum weight, having a pointer, *E*, that traverses a fixed quadrant, *F*, properly divided by a scale to denote the relative pressure exerted through the medium of the receiving pulley on the shaft; the motions are absolute, there being no chance for "backlash."

75. Calibration of a transmitting dynamometer, as an example, one of the Morin type, is illustrated in the following:*

* Distribution of Internal Friction of Engines; R. H. Thurston; Trans. Am. Soc. M. E., Oct. 1888.

The best result in this work was given by such a dynamometer built in the Sibley College

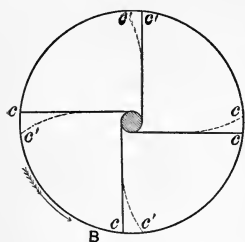


FIG. 110.

shops, at Cornell University. Its action is like that above described and is shown clearly by Fig. 110. A pulley, of which the rim, *B*, is shown, is fitted loose on the shaft, *S*. Four flat springs are securely bolted to the shaft, *S*, and to the rim, *B*. Now, if force be applied by a belt around *B* to turn the pulley, and if resistance to its turning be produced by a fixed pulley on the shaft, *S*, from which some machine is driven by the belting, the springs, *c*, will be deflected into new positions, *c'*, an amount proportional to the force, and the fixed pulley will then revolve, thus driving the machine. To show the amount of power transmitted, and any variation that may occur in that power, a pencil is attached to the rim of the pulley, or to a post having an equivalent motion, and a recording apparatus, consisting of a series of gear wheels actuated by a spiral thread on a sleeve on the axis, causes a band of paper to move radially under the pencil. The recording apparatus can be stopped or started at will, without interfering with the motion of the machinery, by causing the loose sleeve to engage with a lug on the shaft. The diagrams obtained from the dynamometer consisted of a series of waving

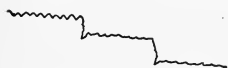


FIG. 111.

lines (Fig. 111) of varying elevation and with different average ordinates. The undulations were produced by changes of speeds probably caused by the inequalities of belt-lacings, etc.

The dynamometer was calibrated in three ways: First, by attaching a brake to the same shaft, and comparing the diagrams with the brake-readings; secondly, by direct pull with a spring-balance against the springs of the dynamometer, and thus obtaining the ordinate for a given belt-pull; thirdly, on the same principle as the first, but a spring-balance was used, to measure the brake-weights, instead of scales. The object

of these calibrations was to obtain the ordinate corresponding to any given belt-pull. The following results were obtained, in the last case, by direct pull against the springs of the dynamometer, which method being employed, gave uniform and satisfactory results.

| Pull on Dynam. Pulley. Pounds. | Ordinate. Inches. | Pull on Dynam. Pulley. Pounds. | Ordinate. Inches. |
|-----------------------------------|-------------------|-----------------------------------|-------------------|
| 0 | 0.40 | 35 | 1.80 |
| 5 | 0.65 | 40 | 2.08 |
| 10 | 0.80 | 45 | 2.32 |
| 15 | 1.02 | 50 | 2.58 |
| 25 | 1.33 | 60 | 3.08 |
| 30 | 1.55 | 70 | 3.52 |

The mean of three results corresponds very closely to the last, and, when plotted, gives a straight line, whose equation is $Y = 0.046X + 0.20$, Y being expressed in inches and X in pounds. According to General Morin, a good dynamometer should have (1) sensibility properly proportioned to the efforts to be measured; (2) the indications should be placed beyond the influence of the observer and given by the instrument; (3) the observer should be able to measure the effect at every point of the curve produced by the machine; (4) the apparatus should be constructed to give the total amount of work.

76. The Prony Brake, the Absorbing Dynamometer, or the Dynamometric Brake, has many forms.—A simple form of dynamometric brake for small powers is illustrated in Fig. 112. A is the shaft of the motor of which the power is to be determined; B is the pulley or drum on which the brake-blocks are changed by the bolts, C, C . The lugs, D, D , limit the movement of the beam, E , which is counterbalanced by a fixed poise at F , and the weights equilibrating the effort of the motor are applied at G .

Basswood and poplar are excellent woods for use in the rubbing parts of the dynamometric brake; but any wood will work well if properly handled. The soft are usually found better than the hard woods, but white ash and maple are good. End-grain is often preferred for rubbing surfaces. The wood

may be secured either to the pulley or to the strap of the brake. Where to be much or continuously used under heavy wear, it is perhaps better to put the blocks on the wheel.

The dynamometric power may be ascertained by means of a *rope-brake* upon the fly-wheel of the engine. Two ropes are

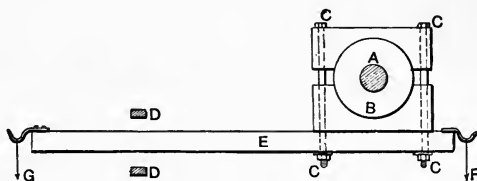


FIG. 112.—DYNAMOMETRIC BRAKE.

used for each wheel, kept at proper distances apart on the wheel by means of transverse wooden distance-pieces. The dead-load is usually applied by means of weights, and the back tensions necessary by means of a spring-balance. The spring-balance tension is deducted from the dead-load applied. This brake is found to work perfectly satisfactorily. If any metal be used for attaching the wooden cross-pieces to the ropes, it must not rub against the rim of the wheel; if this happens, the metal becomes hot, and is liable to burn the rope.

If it is assumed that L = length (effective) of the arm of the dynamometer in feet, W = weight (unbalanced) suspended on the arm in pounds, N = number of revolutions per minute, the horse-power will be

$$\text{D. H. P.} = \frac{2\pi WLN}{33000} = 0.0001904 WLN.$$

It is not unusual to make the effective value of r equal to $\frac{33}{2\pi}$; so that, the circumference described being 33 feet, the power is at once completed by multiplying the weight and number of revolutions of the shaft, W and N , together, and dividing by 1000 to get the horse-power.

A form of this brake which the Author has most frequently employed, and with satisfaction, is constructed as follows:*

Like nearly all dynamometers of this class, it includes a brake-wheel, or pulley, which is keyed on the engine-shaft, and is sufficiently strong to sustain safely the maximum load anticipated. The rim of this pulley is turned flat and smooth, and fitted with a flexible brake-strap of wrought-iron, or other suitable material, which may be adjusted to such a tension as will enable it to control the engine at maximum power. In this case, the rim is trough-shaped in section, flanges extending inward toward the shaft to a sufficient depth to permit the retention in the circular trough so formed of a stream of water which is used to keep the pulley cool, and to carry away the heat produced by transformation of mechanical energy. The two ends of the brake-strap are united by a right- and left-hand screw, in such manner that they may be drawn together and the strap set up to any desired degree of tension. The brake-arms consist of two beams of wood, forming a < frame, and secured to the strap at the upper and lower sides, and at their junction supported by a strut resting on a platform-scale of nice construction and great accuracy. As the engine-shaft revolves, the tendency of the brake-arms to turn is resisted by the scale; and the effort so measured, multiplied by the relative velocity of the engine-shaft and the supported point on the arm, gives a measure of the power expended. Water is supplied to the pulley-rim, by means of a hose, from any convenient source, and the excess is taken away in a similar manner. The centrifugal action of the rotating mass keeps the fluid in place in the pulley-rim, and the eduction-pipe receives the water carried away by it as the tender of a locomotive scoops water from between the tracks, when at high speed. This system permits efficient lubrication, without admixture of grease with the water, and secures a perfection of smoothness and uniformity of rubbing-surfaces unattainable with older forms of brake.

* Construction of a Prony Brake; R. H. Thurston; Journal Franklin Institute, April 1886.

77. Designing a Brake.—The following is an account of the design of a brake, which worked well under higher engine loads than the Author had ever before known to be controlled by this means.* It also illustrates fully its theory.

The brake was designed for the maximum power of the engine, *i.e.* taking steam at full stroke, the engine running at 100 pounds pressure, and at 100 revolutions per minute. The diameter of the cylinder was 18 inches and the stroke 42 inches, and we have for the maximum power developed

$$\text{H. P.} = \frac{254.47 \times 100 \times 42 \times 2 \times 100}{33000} = 540 +$$

The brake was accordingly designed to control the engine when exerting this power, and to be used upon a pulley of 5 feet diameter and 24-inch face. The size of the pulley was chosen of this diameter, simply because it compelled less removal of floor and railings about the engine, and would also cost less than a larger one.

The calculations for the remaining parts of the controlling apparatus is as follows :

Assumed diameter, 5 feet; assumed maximum speed of engine, 100 revolutions; circumference, 15.708 feet. This would give for the greatest linear velocity of the pulley per minute, 1570.8 feet. Dividing the number of foot-pounds developed by the engine at its maximum speed and pressure, by the linear velocity, gives the resistance at the rim of the pulley

$$\frac{540 \times 33000}{15708} = 11345 \text{ pounds;}$$

which figure is the total friction, in pounds, on the face of the pulley.

The brake-blocks were $2\frac{1}{2}$ inches thick, 5 inches wide, and 24 inches long, of unseasoned white-oak, and were placed 7 inches from centre to centre, leaving a space of 2 inches, between adjacent blocks, for diffusion of the heat and for lubrication. The blocks were attached to the flexible brake-straps

* The designers were Messrs. Gately and Kletsch.

by means of wrought-iron lag-screws. The three blocks at the top and those at the bottom of the pulley were fastened to the arms of the brake.

The straps, two in number, were calculated thus:

Let T_1 and T_2 represent the tensions at the ends of the band which embraces the pulley, and let T_1 be the maximum tension.

Then T_1 exceeds the tension T_2 by an amount equal to the friction between the blocks and the pulley; *i.e.*,

$$R = T_1 - T_2 = 11345.$$

Let c denote the ratio which the arc of contact bears to the circumference of the pulley, f the coefficient of friction between the blocks and the pulley; then the ratio $T_1 : T_2$ is the number whose common logarithm is $2.7288cf$; or,

$$\frac{T_1}{T_2} = 10^{2.7288fc} = N.$$

c , the arc of contact of the bands, = 1, and f , the coefficient of friction between wood and cast-iron (well lubricated), was taken at 0.2; then

$$N = 10^{2.7288fc} = 10^{2.7288 \times .2 \times 1};$$

or,

$$\frac{T_1}{T_2} = 10^{0.54576} = 3.5.$$

Having found $R = 11345$ pounds, we have for the greatest tension on the band

$$T_1 = R \frac{N}{N-1};$$

and substituting the values of R and N in this equation, we have

$$T_1 = 11345 \frac{3.5}{3.5-1} = 15883 \text{ pounds.}$$

Hence, for the combined tension on the band, and using two straps, we have for the tension on one

$$\frac{15883}{2} = 7941.5 \text{ pounds.}$$

Taking the tensile strength of such wrought-iron as safe at 40,000 pounds per square inch, and allowing for a sixfold factor of safety, we obtain for the section of the band

$$\frac{7941.5 \times 6}{40000} = 1.19 \text{ square inches.}$$

The nearest band-iron of this section was $\frac{3}{8} \times 3$ inches, and, after careful testing, it was found to be of sufficient strength; giving, at the same time, that flexibility which is of vital importance in the operation of brakes. At each end of the bands it was found necessary to weld on round bar-iron of equal section, to admit of threads being cut for the purpose of tightening and loosening the brake.

The arms were two in number, of 6×6 inches well-seasoned spruce. The length was made 10 feet 6.1 inches from centre of the bearing-surface on the pulley to centre of bearing-surface on the scale, as it brought the scale beyond the rim of the fly-wheel, and also greatly facilitated calculations of the horse-power developed—the circumference of a circle, whose radius is 10 feet 6.1 inches, being 66 feet. Thus instead of multiplying by 66 feet, and then dividing by 33,000 to obtain the horse-power, it is only necessary to divide the product of the net scale-pressure and the revolutions per minute by 550, the quotient being the horse-power developed, *i.e.*,

$$H. P. = \frac{W \times Rev. \times 66}{33000} = \frac{W \times Rev.}{550}.$$

The stand through which the pressure was transmitted to the scale was composed of two uprights, 6×6 inches, of white pine, surmounting a pedestal covering the greater part of the

scale platform. Upon these uprights was placed a steel plate of $\frac{3}{4}$ -inch thickness, which received the pressure of the bolts. The scale was carefully balanced, and was capable of accurately weighing 3000 pounds. All weights used were carefully weighed on a standard balance, and none were used that were found not to be absolutely correct.

As the common segmental arm would give but a very narrow bearing for the rim, the Author advised an arm of I-section, which was found to answer the purpose.

The calculations for the parts of the pulley were made according to Unwin,* giving for the thickness of rim

$$t = 0.7\delta + 0.005D = 0.65 \text{ inch};$$

where D = diameter in inches = 60 inches;
and δ = thickness of belt taken at 0.5 inches.

The number of arms was assumed at 6; and similarly, for the thickness at the nave,

$$h = 0.1781 \sqrt[3]{\frac{PD}{n}} = 8.54 \text{ inches};$$

P being the driving effort, 11345 pounds;

D = diameter = 60 inches; and

n = number of arms = 6;

h_s = breadth of arms = $\frac{h}{2} = 4.27$ inches.

For h at the rim, we take $\frac{2}{3}$ the diameter of the nave.

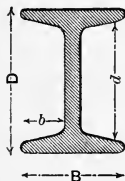
For the thickness of the nave,

$$\delta = 0.18 \sqrt[3]{BD} + \frac{1}{4} = 2.1 \text{ inches};$$

where B is the face = 24 inches;
 D is the diameter = 60 inches.

* Machine Design.

The diameter of the main shaft being 9.12 inches, the calculated thickness of the nave was judged rather small, and 2.5 was used instead. The rim was also made $\frac{1}{8}$ inch heavier at the centre than the calculated dimension used for the edges of the rim.



For the moment of inertia of this section of arm,

$$I = \frac{1}{12}BD^3 - \frac{1}{12}(2bd^3);$$

and, considering the arm as fixed at one end and loaded at the other,

$$Pl = \frac{RI}{d}; \quad R = \frac{Pl d}{I};$$

where

P = load;

l = length of arm;

I = moment of inertia;

R = modulus of rupture;

$d = \frac{1}{2}D$.

Load on 1 arm = $\frac{1}{6}$ of 15600 = 2600 = P .

$l = 30 - 7\frac{1}{2} = 22.5$ inches.

$I = \frac{1}{12}(4 \times 8^3) - \frac{1}{12}(2 \times 1\frac{1}{4} \times 5^3) = 40.6$

Allowing a factor of safety of 8, we have

$$R = 17290,$$

and the above sections and dimensions are ample.

The standardization of this type of absorbing dynamometer consists simply of the careful determination of its exact dimensions, and of the accuracy of the scales employed.

In the original device of Prony, the efforts on the brake were obtained by loading a scale-pan, a far less manageable system than that which is above described. Many modifications of this instrument have been made by various ingenious engineers, some of which may be here briefly referred to.* In all cases and in all applications of the brake, the essential conditions are, mainly, accuracy of measurement and readings, and,

* See Beaumont on Friction-Brake Dynamometers; Proceed. Brit. Inst. C. E., 1888, for a very full account of these instruments.

above all else, uniform friction-resistance, such as can only be insured by good methods of lubrication. All conditions of the trial should be as constant as possible. For small powers, the arm may be omitted and the restraining effort exerted directly upon the strap, and this strap is sometimes merely a leather band or a rope, either with or without blocks. In all cases the work in actual horse-power with the load P in pounds at the distance R in feet from the centre of the wheel, making N revolutions in time t in minutes, will be

$$H.P. = \frac{(R \times 2) \times \pi \times P \times N}{33000 \times t};$$

or taking c = circumference of circle of radius R , and V = velocity in feet per minute of circumference, then $V = cN$, and

$$\begin{aligned} H.P. &= \frac{cNP}{33000t} &= \frac{VP}{33000t}, \\ P &= \frac{33000 H.P.}{cN} &= \frac{33000 H.P.}{V}, \\ c &= \frac{33000 H.P.}{NP}, &N &= \frac{33000 H.P.}{cP}. \end{aligned}$$

In many forms of brake, it is sought to secure a self-adjustment of the grasp of the strap, as, for example, the dynamometer of Messrs. Amos and Appold, Fig. 113. It is provided with a compensating lever, K , by means of which the rise or fall of the load P is attended with a decrease or increase in tension on the brake-strap. With a given tension in the belt, and with the load P set with its point of suspension H opposite the point T , the lever takes a vertical position; but as soon as the load P is lifted the lever pivoted at X moves with and virtually increases the length of the belt, and thus slackens it, allowing the load again to descend. If the total friction decreases, the descent of the load carries the compensating lever

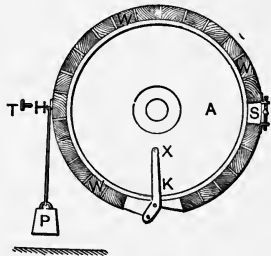


FIG. 113.—COMPENSATING BRAKE.

to a new position, thus tightening the belt and increasing the friction.

In a form of compensating brake designed by Mr. Balk, Fig. 114, the compensating lever is outside the circumference of the strap. It is connected at *B* and at *C* to the strap, and a fixed pin, *F*, passes through a slot at the outer end, where is suspended a weight, *p*, sufficient to keep the lever free. This

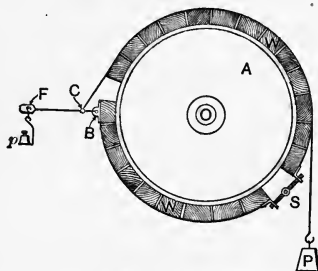


FIG. 114.—COMPENSATED DYNAMOMETER.

weight must be varied with change of condition of the brake-blocks, the lubricant and the temperature of the wheel, and as it must be taken as acting at the radius *OF* in favor of the weight *P*, these variations become troublesome by virtually making *P* a variant.*

The form of pulley which has been described is illustrated in the next figure, which shows the proportions adopted by Mr. Halpin, who was probably the first to use it.

Mr. Beaumont has proposed the following expression as a convenient means of comparing the relative capacity of brakes, judging by the amount of work for which they have been designed, or to which they have been put. The coefficient

$K = \frac{WV'}{HP}$ may be employed, *W* being the width of the wheel

* See paper by Professor Brauer; Zeits. der Ver. Deutschen Ing., Band xxxii; Seite 56.

in inches, and V' = the velocity of the periphery of the wheel in feet per minute. This gives for the

Royal Agricultural Society's single brake $K = 824$

“ “ “ treble “ $K = 495$

Garrett's water-cooled brake $K = 740$ Ransomes, Balk's brake $K = 1020$

Compared in this same manner the Prony brake described by the Author, as devised for measuring a maximum of 540 H. P., the wheel being 5 feet in diameter and 2 feet wide, and $N = 100$, gives $K =$ only 75. This brake was freely lubricated with beef

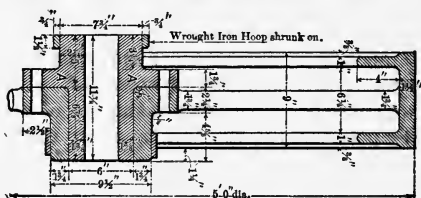


FIG. 115.—BRAKE PULLEY.

tallow, plumbago, and lard oil, and although designed for a maximum of 540 H. P., it was not worked above 200 H. P., and at this power $K = 188$. The more effective the brake the lower the coefficient. $K = 400$ may be taken as a good figure; the coefficient is less, apparently, on the water-cooled hollow rim. Other things equal, the highest coefficients correspond to largest areas of rubbing-surface on the rim. Mr. Beaumont makes the following approximate estimate of the maximum pressure p per square foot of block. W' being the width of the wheel in feet, p may be obtained thus:

$$T + T = DpW', \quad \text{or} \quad T = \frac{DpW'}{2},$$

and

$$p = \frac{T + T}{DW'}.$$

The greatest pressure per square foot of surface of the Author's example of a Prony brake, if it had been used as proposed for 540 H. P., would have been, at $f = 0.2$,

$$p = \frac{15883 \times 2}{5 \times 2} = 3176 \text{ lbs.}$$

At 200 H. P. $p = 1180$ lbs. The greatest pressure per square foot on the blocks of the Royal Agricultural Society's brake at Newcastle was $p = 312$ lbs. with the blocks covering 0.8 of the surface of the wheel.

This is a rough approximation to the greatest pressure per square foot, but it affords a fair means of comparison of the pressure with the different brakes.

A good method of fitting the dynamometric brake to a portable engine is shown in the illustration.* A is the beam of the brake; B is the point of attachment of a strut which transmits the pressure due the turning effort to the platform-scale, seen below; C is a counterbalance.

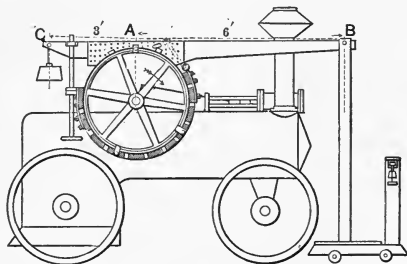


FIG. 116.—DYNAMOMETRIC BRAKE.

In handling a brake cooled by water, it is advisable to use as small a quantity of the liquid as possible, and it will often be found perfectly practicable to even permit its complete evaporation, keeping the metal at 212° F., thus reducing the

* On the Distribution of Internal Friction of Engines; R. H. Thurston; Trans. Am. Soc. M. E., 1888.

quantity demanded to a small fraction of that often used, and largely doing away with the difficulties incident to its supply and wholly with those of discharge.

In the Brauer dynamometric brake,* Fig. 117, instead of wooden jaws hugging the rims of the pulley, an iron band is used for flat-rim pulleys, and wire-ropes for grooved pulleys. The apparatus is composed of the following parts:

1st. The iron band or wire-rope.

2d. The clutch-producing arrangement and the regulation of the tension of the clutch-producing bands or ropes.

The iron band applied to an ordinary flat-rim pulley is provided with four double guides k (Fig. 117) to retain the band in position, fixed up by means of a stirrup bolt and link and a safety cord $l'l'$, so as to allow the band to have a play of 100 millimetres. These safety measures can be modified according to circumstances.

The frictional parts are actuated in the following manner:

The wire band is attached at its ends to the point of the application of resistance a , and to the point of rotation b by a lever abc . The point c of this lever is attached at d by means of a winding tackle ee_2 , and a spring f , to the upper end of the wire band. The cord of the winding tackle after leaving e is passed round the friction-roller g . The operator, by pulling the cord, produces the tightening action in such a way that the weight p is lifted and equilibrium established. In order that this condition may not be affected by the tension of the part of the cord from e_2 to g , it is only necessary to fix the friction-roller in such a way as to be in line with the axis of the pulley or drum. The moment of this tension will then be *nil*. The friction-roller is not indispensable, but its application permits the operator to control the action of the brake at some distance. Automatic regulation is effected, according to the variations of the friction, by the combined action of the spring f and the cord h . It will be seen that if, owing to excessive friction, the weight p is lifted above its mean position, the cord

* "Bulletin de la Société Industrielle de Mulhouse," 1884, p. 485. Proc. Inst. C. E., No. 2079.

h will be stretched, augmenting the tension, and elongating the spring f . The result is a diminution of the clutching action, and the weight p will be lowered.

The weight l , which obviates the necessity of attaching the cord h to the floor-boards, should be equal the strain liable to be brought upon the cord h .

The lifting-tackle should always be suspended in such a way as to reduce the traction upon the cord h to a minimum, and so that the influence of this factor upon the condition of equilibrium in general need not be taken into consideration.

Lubrication may be effected in any convenient manner. The rim is usually sufficiently cool under light load without water.

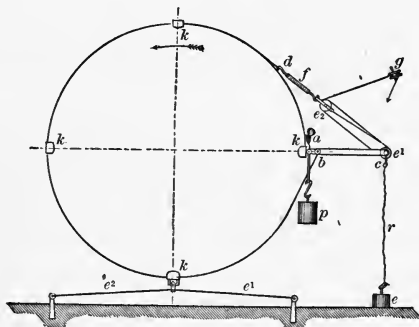


FIG. 117.—BRAUER'S BRAKE.

78. Data obtained by the use of the transmitting dynamometer are often of great value as a check on other methods of test. Mr. Emerson gives the following as illustrating this fact:

Testing woollen-mill machinery, running ten "sets," eleven hours per day, with the boiler-pressure kept at 70 pounds, the driving-pulley on engine, 9 feet diameter, with 30-inch double belt, drove a 5-feet pulley upon the main line. Throwing on and off machinery caused a variation of four revolutions of the pulley on the engine, or from 120 down to 116 per minute, $3\frac{1}{4}$ per cent.

The results obtained were as follows:

Average net effort for 11 hours, 1163 lbs.
 Coal burned in 11 hours, 4955 lbs.
 Average power in 11 hours, 82.9 H. P.
 $4955 \div 11 = 450.4 \div 82.9 = 5.43$ lbs. coal per horse-power per hour.

The St. Joseph Milling Co., Mishawaka, Ind., 100-barrel mill, required 3.81 horse-power per bushel.

The Ripple Mill, Mishawaka, Ind., 130-barrel mill, required 3.91 horse-power of water per bushel ground.

Mishawaka Mill, Mishawaka, Ind., 175-barrel mill, required 4.72 horse-power of water per bushel.

Sage Brothers' Flouring Mill, Elkhart, Ind., roller-mill, 280-barrel capacity, required 3.18 horse-power of water per bushel ground.

TESTS ON SPINNING FRAMES, SPEEDERS, ETC.

| No. OF TEST. | Time. | No. of Spindles. | No. of Yarn. | No. of Weighings. | CONDITION. | No. OF TURNS PER MINUTE OF SPINDLES. | | | | No. OF TEST. |
|--------------|------------------|------------------|-----------------|-------------------|----------------------------------------------|--------------------------------------|-----------|------------|----------|--------------|
| | | | | | | Heaviest. | Lightest. | Variation. | Average. | |
| 1 | 2.00 till 5.30 | 192 | 28 ³ | 6 | Bobbins from $\frac{1}{2}$ full till doffed. | 7343.5 | 7556.0 | 212.5 | 7449.2 | 1 |
| 2 | " " | " | " | 3 | Bobbins from doff till $\frac{1}{2}$ full. | 7392.0 | 7343.5 | 9.3 | 7419.3 | 2 |
| 3 | 8.00 till 10.00 | 192 | 28 ³ | 7 | Total of above. | 7343.5 | 7343.5 | 0 | 7434.4 | 3 |
| 4 | 10.30 till 11.30 | 192 | 28 ³ | 6 | Bobbins full till $\frac{1}{2}$ full again. | 7228.5 | 7362.8 | 134.3 | 7287.2 | 4 |
| 5 | | | | | Bobbins $\frac{1}{2}$ full till full. | 7355.4 | 7246.8 | 108.6 | 7324.2 | 5 |

| No. OF TEST. | HORSE-POWER TO A FRAME. | | | | SPINDLES PER HORSE-POWER. | | | | No. OF TEST. |
|--------------|-------------------------|-----------|------------|----------|---------------------------|-----------|------------|----------|--------------|
| | Heaviest. | Lightest. | Variation. | Average. | Heaviest. | Lightest. | Variation. | Average. | |
| 1 | 1.7000 | 1.6150 | .0850 | 1.6660 | 113.0 | 119.0 | 6.0 | 115.3 | 1 |
| 2 | 1.5050 | 1.3820 | .1230 | 1.4420 | 127.7 | 138.0 | 0.3 | 132.4 | 2 |
| 3 | 1.7000 | 1.3820 | .3180 | 1.5915 | 113.0 | 138.0 | 25.0 | 120.7 | 3 |
| 4 | 1.3504 | 1.2500 | .1004 | 1.3106 | 142.2 | 153.6 | 11.4 | 146.5 | 4 |
| 5 | 1.6180 | 1.4205 | .1975 | 1.5094 | 118.6 | 135.0 | 16.4 | 127.2 | 5 |

Mr. S. Webber finds the friction of this class of dynamometers to be sensibly constant.

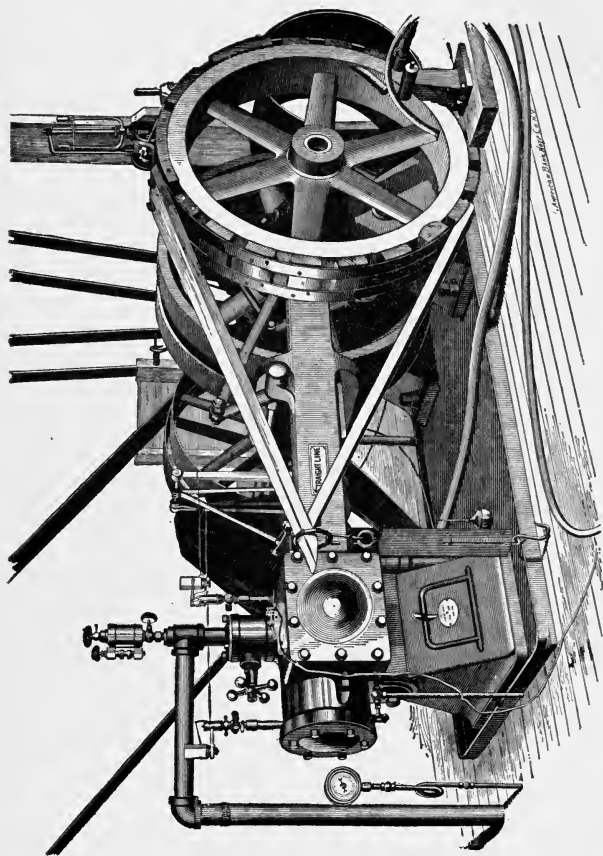


FIG. 118.—FITTING UP FOR TRIAL.

CHAPTER VIII.

STANDARD METHODS OF ENGINE TRIAL.

79. Standard Methods of Engine Trial have been proposed by various writers and practitioners, with the double purpose of securing all needed data, at least cost in time and money, and of making all results strictly comparable. In the absence of agreement in regard to method, a great variety of practice is liable to spring up, and as great a variety of methods of securing, tabulating, and computing results from the data obtained. It is therefore considered advisable that all such experimental work, whether for directly practical purposes or with a scientific object, should be made by that carefully planned and precisely stated system, which should be best adapted to the ready determination of all needed data, with least liability to error, and a most convenient means of checking all figures. Both engine and boiler trials should be made in accordance with such a system as is generally recognized as well adapted to its purposes, and accepted and indorsed by those whose learning and professional standing and experience best fit them to judge it. Standard engine and boiler trials are not yet as widely adopted as is desirable; but they are gradually taking definite form and are steadily coming into general use.

The results of any engine trial, if complete and accurate, should enable the engineer to answer several questions:

(1) What is the real efficiency and the economical performance of the system tested?

(2) How does it compare with standard apparatus of a similar character? and in what is it superior or inferior? What are its excellencies and its defects?

(3) How are commercial and financial conditions affected by its operation?

These questions being solved, the proprietor will know to what extent his expenditures and his methods of operation are wise and productive; the builder will learn how successful he has been in his work, where it is defective, and what remedy is available; the engineer secures data which enable him to design intelligently later and better constructions, and it may furnish a standard for still other comparisons.

The duty-measurement should always be expressed in perfectly definite terms. The usual expression may be interpreted to assume any one of several different units. If the efficiency of the system, engines and boilers included, is to be measured only, it is sufficient to ascertain the relation between the work performed and the cost of its performance as measured at the boilers; but even this may be a very uncertain measure unless the quality of the fuel is prescribed. The kind of fuel should be stated in all such cases; but a system of measurement which determines the heat produced in the furnace, in thermal units, or other equally definite terms, and the quantity of *useful* work which it yields, is the only satisfactory one.

The only correct and exact method of gauging the performance of any steam-engine is to determine the weight of steam or, better, the number of thermal units demanded by it per horse-power per hour. The proper measure of the boiler efficiency is the proportion of the heat of combustion of the fuel which is absorbed and stored as available energy in the steam which it produces. To rate the engine by the quantity of fuel burned at its boiler is wholly incorrect; to rate the boiler by the ratio of steam made to coal burned is hardly less indefinite. It is only by the habitual use of a known fuel of uniform composition and physical character that comparisons of value may be effected at all. Even where the steam-unit is adopted, it must be taken at a standard temperature and pressure. In all heat-engines the proper measure of heat-energy is the heat-unit. There is, therefore, reason in the adoption of, for an exam-

ple, 1000000 B. T. U., a figure sometimes so taken, as a standard quantity in duty-trials of engines.*

80. Engine and Boiler Trials are so generally conducted together, and the former so commonly depend for their essential data upon the latter, that the presentation of illustrations of standard methods for both is considered advisable and will be given later. Of the two, the latter is usually much the more laborious and troublesome, and also more expensive in time and money. The boiler-trial also admits of a greater variety of methods than the engine-trial, and is correspondingly more liable to yield results of varying accuracy. For this case the best practice is better settled than for the other, and standard methods of steam-boiler trial are fairly well established on both sides the Atlantic.

A good boiler should have an efficiency of not less than 75 per cent., giving thus 10875 B. T. U. per pound of carbon supplied as fuel, and, deducting ash, for good coal, about 10000 B. T. U. per pound of fuel, the equivalent of 0.25 pound of coal per horse-power and per hour, if all converted into work. The real efficiency of any engine is measured by the quotient of this quantity for the actual consumption. Thus, an engine using 2.5 pounds of good coal has an efficiency of heat

$$E = \frac{0.25}{2.50} = 10 \text{ per cent.}$$

This is considered a good result. Ninety per cent. of all heat supplied is here wasted. An engine using 1.25 pound of best fuel, a result sometimes claimed, but certainly seldom reached, has an efficiency of

$$E = \frac{0.25}{1.25} = 20 \text{ per cent.}$$

A consumption of 40 pounds of steam would give

$$E = \frac{2.5}{40} = 6 \text{ per cent.}$$

* This corresponds to 100 lbs. coal evaporating 11.25 lbs. water from and at 212° F.

As an example of a systematic scheme of engine-trial, illustrating the general characteristics of a standard method, we may take the following outline abstract from a plan for a pumping-engine duty-trial proposed by Mr. Barrus : *

(1) It is presumed at the outset that the engine is in thorough order in every part, having been in operation a sufficient length of time, since its erection, to secure easy and proper working. If this has not been done, and especially if the plunger has been recently packed or its packing newly re-adjusted, the engine is worked for a run of at least twelve hours' continuous service in preparation for the test.

(2) The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of at least three hours, so as to find the temperature of the feed-water (or the several temperatures, if there is more than one supply), for use in the calculation of the duty. During this run the observations are made every fifteen minutes, and the results averaged.

(3) The engine is now stopped for a time, in order, first, to connect up the measuring apparatus for determining the weight of the feed-water consumed, or of the various supplies of water if there are more than one ; and, second, to test the leakage of the plungers.

The quantity of water which leaks by the plungers is most satisfactorily determined by removing the cylinder-heads. A wide board or plank is temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water, in the manner of a dam, and an opening is made in the temporary head, thus provided, for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or if this position is not practicable, at the end of the stroke), and the water from the force-main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured. The test need not continue over fifteen minutes, or, if carefully made, a less time, the desired object being to get a satisfactory deter-

* London Engineering, Mch. 1, 1889, p. 217.

mination of simply the rate of leakage. If no means exists for putting the back side of the plunger under water-pressure, a suitable pipe can readily be provided for the purpose. Should the escape of water in the engine-room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the temporary wooden head. The apparatus may be constructed in a somewhat rude manner, and be sufficiently accurate for practical requirements.

In the case of a pump from which it is difficult to remove the cylinder-head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction-valves, the head being allowed to remain in place. If the test is made without removing the head, leakage of the discharge-valves may be confounded with leakage of the plunger. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger-test. The discharge-valves on the back end of the pump should likewise be examined, as also the suction-valves on both ends, and the disordered valves removed. Leakage of the discharge-valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction-valves will be shown by the disappearance of water which covers them.

The leakage-test being completed, no change is allowed in the adjustment of the packing of the plunger (supposing this to be of a form capable of adjustment), the head is immediately replaced and preparations made for at once beginning the main duty-trial.

(4) The duty-trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler, as well as that of the engine. The test of the two will go on simultaneously after both are started, although the boiler-test will begin a short time previous to the commencement of the en-

gine-test, and continue after the engine-test is finished. The mode of procedure is as follows :

While the preparations are being made to start the engine, after the completion of the leakage-trial, steam is raised in the boiler to the working pressure. The fire is then hauled, the furnace and ash-pit cleaned, and the test of the boiler is commenced. This test is made in accordance with the rules for a standard method recommended by the Committee on Boiler-Tests of the American Society of Mechanical Engineers.* This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler being previously heated to its normal working degree; operating the boiler in accordance with the conditions determined upon, weighing coal, ashes, and feed-water; observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam: and at the close of the test hauling the fire and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam-pressure being the same as at the time of lighting the fire at the beginning of the test. The temperature of the feed-water is observed at the point where the water leaves the engine-heater if this be used, or at the point where it enters the flue-heater if this apparatus is employed. In either case, where an injector is used for supplying the water, a deduction is to be made for the increased temperature of the water due to this method of feeding.

As soon after the beginning of the boiler-test as practicable, the engine is started and preparations are made for the beginning of the engine-test. The formal commencement of this test is delayed till the plant is in normal working condition, which should be not over one hour after the time of lighting the fire. When the time for commencement arrives, the feed-water is momentarily shut off, and the water in the lower tank

* See this report as given in Chapter II of the present work.

is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass, and the indication of the counter on the engine, after which the supply of feed-water is started and the regular observations of the test commenced. The test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter observed, together with the number of tanks of water thus far supplied, the engine-test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling and completing the boiler-test. It is then stopped, and the final observations relating to the boiler-test are taken.

The observations to be made and data obtained for the purposes of the engine-test embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner already pointed out. They also embrace the number of hours' duration and number of strokes of the pump during the test, as noted, together with the length of the stroke (in direct-acting engines), the indication of the gauge attached to the force-main, and indicator-diagrams from the pump. It is desirable that indicator-diagrams be obtained also from the steam-cylinders.

Observations of the length of the stroke should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments—such as steam-gauge, vacuum-gauge, thermometer in pump-well, thermometer in feed-pipe, thermometers showing temperature of engine-room, boiler-room, and outside air, thermometer in flue, thermometer in steam-pipe if the boiler has steam-heating surface, barometer and other instruments which may be used—every half-hour; indicator-diagrams should be

taken every half-hour, both from the steam and from the water cylinders. Should the diagrams from the pump be rectangular, they may be taken, if desired, with less frequency.

When the duty-trial embraces simply a test of the engine apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled and the special observations relating to the performance of the boiler will not be taken.

(5) In making preparation for the test, attention should be given to the following provisions in the arrangement of the apparatus:

The gauge attached to the force-main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force-main. By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the centre of the gauge, and correction for this element of variation avoided.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, this method of measurement is preferred. The personal errors in observing the exact scale-marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used which acts directly on the moisture tested. If

either the superheating calorimeter or the wire-drawing instrument be employed, the steam which it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface-condenser of some simple construction, and measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations, the specific heat of the material of the barrel should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam-pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator-pipes and connections used for the water-cylinders should be of ample size, and so far as possible free from bends; $\frac{3}{4}$ -in. pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator-springs which are correct under steam heat are erroneous when used for cold water. When steam springs are used, the amount of error should be determined if calculations are made of the indicated work done in the water-cylinders.

To avoid errors in conducting the test due to leakage of stop-valves either on the steam-pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

(6) The engine is to be worked on the duty-trial, unless otherwise stipulated, at its rated capacity of discharge.

(7) In review of the method thus pointed out, the various steps may be summed up as follows:

- a. Preliminary run to determine the temperature of the feed-water;
- b. Erection of weighing apparatus, examination of pump, and test of plunger leakage;
- c. Commencement of boiler-test;
- d. " " engine-test;
- e. Boiler and engine test go on simultaneously;
- f. Close of engine-test;
- g. " " boiler-test.

(8) It is desirable that the report of a duty-trial should be sufficiently full to show the performance of the engine and its various members in all other respects than the simple expression of the amount of duty performed. For this reason the horse-power developed by the steam-cylinders, the feed-water consumption per horse-power per hour, the steam accounted for by the indicator, and other information relating to the work of the engine in the capacity of a steam-engine, should be determined and given.

The efficiency of the mechanism of the engine should also be determined and stated, that is, the proportion which the work done upon the water bears to the work done in the steam-cylinders. This efficiency may be expressed by any formula in which the numerator is the duty and the denominator is the work done during the trial, measured from the indicator-cards taken from the steam-cylinders. This efficiency measures a quantity which is of primary importance in the operation of the engine and should always be carefully and exactly determined.

81. Fitting of the Engine for a Test, whether of efficiency or of capacity, is best done in advance of the trial, and ample time should be taken to see that not only all apparatus, but the engine itself, is in readiness; though, if the intention is, as is sometimes the case, simply to ascertain the condition of the engine as found, no other preparations are permissible than those customary before starting up. A good

example of the fitting up of a small high-speed engine is illustrated in Fig. 118, and, in plan, in Fig. 119, in which *AA* is the engine, *BB* its shaft, *CC* the indicators, *D* the indicator reducing-gear, driven from the crosshead; *EE* is the Prony Brake, receiving its cooling water at *F* and discharging it at *G*, and attached to the platform-scale arranged at *H*; the screw tightening its strap is at *I*. The speed-indicators were, in this case, of several kinds. Hand instruments of various kinds were used to check the records of

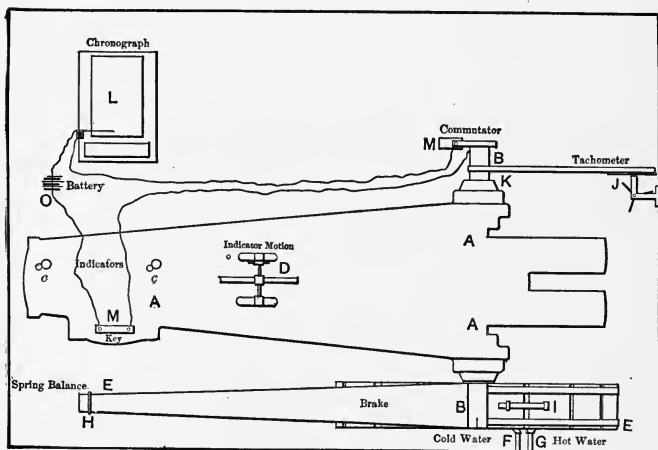


FIG. 119.—FITTING UP THE ENGINE.

the automatic instruments. A "tachometer," *J*, was attached and belted at *K* to the engine-shaft, and afforded a very convenient means of watching the momentary fluctuations due to variations of load, of steam-pressure, and of accidental disturbances. A chronograph at *L* was also attached, connected with a standard clock to beat seconds, and a current was derived from the battery at *O*. A commutator, *M*, was placed on the engine-shaft, making contact at each revolution, and a key, *N*, near the engine, for the purpose of breaking contact. A Brown

mercury speed-indicator served excellently well for a constant speed-indicator. The chronograph was set in operation when the indicator-cards were taken, and thus gave the exact speed of the engine at that instant. Great care must be taken to keep the instruments, and the engine as well, in good order and well lubricated throughout the series of experiments.

82. Two Methods of Trial are available in testing steam-engines, both of which are found to be capable of giving exact results: (1) Measuring the energy supplied by the boiler in the form of heat transferred to the engine by the steam, and comparing the mechanical equivalent of this heat-energy with the quantity of mechanical energy obtained from the engine. (2) Determining the amount of energy rejected, as measured in the heat carried away by the exhaust, and similarly comparing this with the work done. In the first case, the quotient of the useful energy gained by the total energy expended is a measure of the efficiency of the system; in the second case, the same measure is obtained by dividing the work done by the sum of that quantity and the rejected energy. Of these two systems of trial, the first is that customarily employed by engineers for many years past; the second is that comparatively recently introduced by Messrs. Farey and Donkin. Both are fully described elsewhere.

The first system being adopted, the quantity of heat-energy expended is measured by determining the weight of steam produced and its physical condition, and the quantity of heat brought to the boiler by the feed water. The total heat communicated to the steam, less the heat received with the feed, is the net expenditure. It is usual to take a standard temperature at 0° F., 32° F., or 0° C., as that to which all temperature measurements are referred. In such case, assuming the standard point on the scale to be 0° , the total heat supplied by the boiler is ascertained by weighing the feed-water for a specified time, and thus determining the weight of steam, wet or dry, passing to the engine; next ascertaining what proportion of the fluid is still liquid, or what is the amount of superheating; computing the heat stored in the fluid; then, finally, deducting the

| | |
|-------------------------------------------------|------|
| Heat from boiler to engine, per revolution..... | 1377 |
| “ “ “ “ jackets, “ “ | 212 |
| “ “ “ total B. T. U. per revolution..... | 1589 |
| “ returned to boiler, “ “ “ | 38 |
| “ <i>net</i> supply “ “ “ | 1551 |
| “ converted into work “ “ “ | 227 |
| “ total rejected “ “ “ | 1324 |

The loss by conduction and radiation, externally, was about 6 per cent. The actual efficiency of the engine was

$$E = \frac{227}{1551} = 0.146,$$

or not quite 15 per cent., while the thermodynamic efficiency was 0.335, more than twice as great.

This latter method is known as that of Messrs. Farey and Donkin.

In duty-trials of pumping-engines, the best system yet proposed is probably that already mentioned, which bases the efficiency determination upon the measured amount of work done by the system on a consumption of 1000000 B. T. U. supplied in the boiler-furnace, or used in the engine, as the case may be. The heat consumed should be taken to be all supplied by the fuel, or all received by the engine, including that wasted by all its accessories.

The useful work should be, wherever practicable, measured by the product of weight of water pumped, as ascertained by the use of a weir, into the head against which it is pumped, as measured by a pressure-gauge, or otherwise, at the pump-delivery. Losses by leakage, lost action, etc., are thus detected. Internal friction thus properly tells against the engine; external friction—in mains, etc.—is as properly ignored.

83. The Farey and Donkin System of trial of engines is one in which the quantity of heat supplied by the boiler and received by the engine is not directly determined, but is ascertained by observation of the quantities of heat rejected by the engine and carried away in the condensing water. This

method only applies to condensing-engines and to those which can be temporarily converted into condensing-engines for the purposes of the test. A boiler-trial is always a troublesome and disagreeable operation, and usually involves considerable expense both in preparation and in its conduct. Where it is only the engine that is to be tried and judged, the avoidance of a boiler-trial is a decided advantage. The ability to test an engine by itself is very often an important desideratum, and especially as permitting more frequent determinations of the condition of the machine and a more complete knowledge of its action at all times.

It has been seen that the heat supplied to any engine is disposed of in three directions: by conduction and radiation to surrounding objects; by conversion into mechanical work, and by rejection in the exhaust steam and the water accompanying it. Of these quantities the first is comparatively small, and is often entirely ignored as unimportant; the second ranges in good engines between, perhaps, 10 and 15 per cent., rarely exceeding the latter figure; while the last item includes, as a rule, above 85 per cent., and generally 90 per cent., of the total quantity sent over from the boilers. In the condensing-engine all this heat may be found and measured up in the water passing out at the delivery-pipe from the hot-well. It is obvious that the sum of the heat-equivalent of the indicated power of the engine, plus the heat so rejected, and the small quantity added to represent losses by radiation and conduction, will be the measure of the heat-supply from the boiler. To determine this total, therefore, we have but to measure the indicated power of the engine and the heat discharged from the condenser. The first of these processes is already understood. To secure the second measurement, it is only necessary to measure the flow of the heated water by a weir and notch, at the same time measuring its temperature by accurate thermometers. A high value for the quantity of heat discharged, per horse-power and per hour, indicates an inefficient engine; a low value is the proof of good economy.

The apparatus employed by Messrs. Farey and Donkin con-

sists, Fig. 120, of a simple measuring-box, *AA*, of convenient size, six or eight feet long usually, three to five feet wide and two to four feet deep, fitted with a notch, *D*, which is commonly about 6 inches wide. On the engraving this box is of iron, but it is perhaps oftenest of wood.*

It is fitted with transverse partitions *BB*, while at *D* a thin brass or copper plate has formed in it the notch producing the tumbling-bay. The notch in end of the box is larger than the notch in the plate, so that the approach of the water may not be interfered with. The box also is so placed that the water has a clear fall of 12 or 18 inches. The water from the hot-well is delivered into the box at one end, and flows over, under, and through the partitions, as shown, so as to be thoroughly mixed and the current steadied.

The box is provided at *C* with a standard fixed to the bottom, having a hole in it which receives the stem on which the float *e* moves loosely. At the top is a scale of inches capable of being adjusted by screws, while the float carries a pointer *f* which moves up and down this scale with the float.

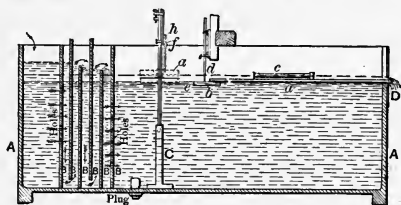


FIG. 120.—FAREY AND DONKIN'S APPARATUS.

To fix the zero-point, a straight-edge *a* is provided, and another, *b*, forming an extension of the bottom edge of *a*; *a* is then placed with one end resting on the notch, while *b* is beneath the gauge *d*, this gauge being free to move vertically in its holder; *a* is then adjusted until the spirit-level *c* shows it to be level when the gauge *d* is fixed by its clamping-screw. The straight-edge and spirit-level are then removed. A scale can

* London Engineering, Feb. 5, 1875.

now be fixed so that its zero agrees with a mark on the gauge stem. If a float is to be employed, the gauge is only used to determine the zero of the float-scale. The float having been put in place, water is admitted into the box until the surface is found just to touch the point of the point-gauge, and the scale is adjusted so that the zero-point agrees with the index of the float.

The depth of water in the notch being measured by the gauge or the float, and the width of the notch being exactly known, the quantity of water flowing is at once readily computed by use of the standard formulas for flow, through a notch or over a weir. The temperatures of the water being taken at the same time, before the water enters and as it leaves the condenser, the product of the mean weight of water flowing per hour by the mean range of temperature measures the heat-units discharged. This quantity, divided by the mean indicator horse-power for the same period, gives the desired figures per indicator horse-power per hour, or

$$H' = \frac{V \times D \times S \times (T_1 - T_2)}{I. H. P.};$$

when H' = heat-units as above ;

V = volume of water flowing per hour ;

D = density of water at the observed temperature ;

S = specific heat, usually taken as unity ;

T = observed temperature ;

$I. H. P.$ = indicated horse-power ;

The quantity H' is often called the constant for the engine.

Since each horse-power demands

$$\frac{1980000}{772} = 2564.5, \text{ or } \frac{33000}{772} = 42.75,$$

heat-units per hour or per minute, the quantity of steam supplying the heat converted into work per hour is

$$\frac{2564.5}{h - t} = w,$$

when h and t are the total heat of the steam and the temperature of the condenser, ranging from $2\frac{1}{4}$ to $2\frac{1}{2}$ pounds, according to circumstances. The heat discharged, H' , being given, the weight of steam supplying it is

$$\frac{H'}{h - t} = w',$$

varying from about 15 pounds upward per *I. H. P.*, the total of both items thus measuring the demand on the boiler, amounting to $w + w' = 17$ or more pounds per horse-power and per hour in good engines. If the boiler "foams" or "primes," this expenditure of feed-water is correspondingly increased. This relation being established, any variation in it or in the "constant" for the engine, as shown at the "tumbling-bay" or weir-notch, indicates some change in the working of the engine, and will call for attention.

Mr. Donkin gives the following table for use in making trials by the Farey and Donkin method :

WEIGHT OF WATER THAT WILL FLOW OVER A TUMBLING-BAY SIX INCHES WIDE.

| Inches over Bay. | Pounds of Water per Minute. | Inches over Bay. | Pounds of Water per Minute. | Inches over Bay. | Pounds of Water per Minute. | Inches over Bay. | Pounds of Water per Minute. |
|------------------|-----------------------------|------------------|-----------------------------|------------------|-----------------------------|------------------|-----------------------------|
| $1\frac{1}{2}$ | 274 | $2\frac{8}{8}$ | 547 | $3\frac{1}{2}$ | 874 | $4\frac{1}{8}$ | 1250 |
| $1\frac{9}{16}$ | 292 | $2\frac{7}{8}$ | 568 | $3\frac{5}{16}$ | 900 | $4\frac{9}{16}$ | 1279 |
| $1\frac{5}{8}$ | 310 | $2\frac{3}{2}$ | 589 | $3\frac{8}{8}$ | 926 | $4\frac{1}{4}$ | 1306 |
| $1\frac{11}{16}$ | 327 | $2\frac{9}{8}$ | 612 | $3\frac{7}{16}$ | 951 | $4\frac{5}{16}$ | 1336 |
| $1\frac{3}{4}$ | 345 | $2\frac{3}{4}$ | 634 | $3\frac{3}{2}$ | 977 | $4\frac{8}{16}$ | 1365 |
| $1\frac{7}{8}$ | 365 | $2\frac{11}{8}$ | 657 | $3\frac{9}{16}$ | 1003 | $4\frac{7}{16}$ | 1394 |
| $1\frac{1}{2}$ | 383 | $2\frac{5}{4}$ | 680 | $3\frac{5}{8}$ | 1030 | $4\frac{1}{2}$ | 1424 |
| $1\frac{5}{8}$ | 402 | $2\frac{13}{8}$ | 704 | $3\frac{11}{16}$ | 1056 | $4\frac{9}{16}$ | 1454 |
| 2 | 421 | $2\frac{7}{4}$ | 727 | $3\frac{3}{4}$ | 1083 | $4\frac{8}{16}$ | 1483 |
| $2\frac{1}{16}$ | 442 | $2\frac{15}{8}$ | 751 | $3\frac{13}{16}$ | 1112 | $4\frac{11}{16}$ | 1514 |
| $2\frac{1}{8}$ | 462 | 3 | 775 | $3\frac{5}{8}$ | 1139 | $4\frac{1}{2}$ | 1544 |
| $2\frac{3}{16}$ | 483 | $3\frac{1}{8}$ | 800 | $3\frac{11}{8}$ | 1166 | $4\frac{3}{8}$ | 1575 |
| $2\frac{1}{4}$ | 503 | $3\frac{1}{4}$ | 825 | 4 | 1193 | $4\frac{7}{8}$ | 1605 |
| $2\frac{5}{16}$ | 525 | $3\frac{3}{8}$ | 850 | $4\frac{1}{8}$ | 1221 | $4\frac{15}{16}$ | 1635 |

N.B.—10 pounds of water is taken equal to one gallon.

84. Trials of Gas-Engines usually involve the determination not only of the indicated and dynamometric power, and the quantity of gas consumed as working fluid and in ignition, but also, if satisfactorily complete, the extent and the method of the several wastes, as by the water-jacket, by conduction and radiation within the working cylinder, and by the exhaust-pipe. The volume of water flowing through the jacket and its variation of temperature readily determine the waste by the jacket, but the measurement of the loss at the exhaust is less easy. This involves the measurement of the volume and density of the gases entering and leaving the engine, and their alteration of temperature. It has been found that, to determine this quantity of fluid, it is necessary to use a meter on both the air and the gas entering the working cylinder of the engine. It is not certain, in any case, that the total volume can be ascertained by the measurement of the cylinder; although, in ordinary work, it may be so taken with a fair degree of approximation to accuracy. The quality of the gas should also be carefully ascertained by analysis. A good engine, using good gas, in sizes exceeding ten actual horse-power, should not consume above 20 cubic feet (566 litres) per indicated horse-power per hour, or 30 feet (850 litres) per dynamometric H. P.; but illuminating gas often may give a result less satisfactory by ten per cent. or more. Many gas-engines of less perfect construction demand double this quantity and upward. A cubic foot of good gas should supply about 620 B. T. U. of heat; a cubic metre should yield about 5600 calories. The theoretical mixture will be usually found to be not far from seven volumes of air to one of gas; but it is better to use a slight excess of air.

As has been already seen, the complete indicator-card is, in the compression type of engine, only obtained after four revolutions, and the observer should be careful to see that he has the diagram of the complete cycle before removing the pencil from the paper.

The data are complete when they permit the computer to show precisely how much gas, how much heat, and how much

energy are supplied to the engine ; how much is applied usefully ; how much is wasted and what are the measures in detail of all wastes ; securing results in such manner, that it is made possible to construct an account that shall exactly or approximately exhibit on its balance-sheet all receipts and expenditures and an exact balance.

85. Simple and Binary Vapor-engine Trials involve no peculiar methods or operations. In these, as in all other cases, the problem of the engineer is the determination of the heat-energy developed and applied in the engine and of the nature and magnitudes of all wastes. In vapor-engines, as in the ammonia or the naphtha-vapor engine, the only differences in its working, when compared with the steam-engine, are due to peculiarities of physical properties, and involve no essential modification of the method of heat or power measurement. The purpose of the trial is commonly to obtain a comparison of efficiency with that obtainable under similar circumstances, with a steam-engine of equally good design and construction, or of standard make and operating in the customary manner. A common practice, on the part of promoters of new schemes in this direction, is to exhibit a comparison with a comparatively wasteful and badly constructed steam-engine. The engineer making such trials should be especially careful in this matter.

The Binary Vapor-engine is commonly a complex machine, composed of a steam-engine and a simple vapor-engine utilizing the heat of the rejected steam. This combination is tested with the steam-engine of fairly comparable design and construction. As a rule, the comparison lies between the combined motor and a condensing steam-engine. The trial determines the efficiency of the steam-engine and the vapor-engine, separately and combined, and should give complete data relating to quantities of heat transferred and transformed, of fuel, and of fluids employed, and of work, useful and lost, as well as of the power developed by each. When testing other vapors than steam, it is often important that their essential chemical and physical characteristics should be redetermined for the occasion ; as they sometimes vary somewhat, as in the case of the petroleum

vapors, and they may differ from the recorded data of the treatises taken as authority.

86. Gas and Vapor Engine Trials may thus demand special treatment in some cases. The specific heats of gas-mixtures may require to be determined; the specific heats of vapors may be undetermined, or their recorded values may be inaccessible. In such cases it may become the duty of the engineer to ascertain their values by computation or by direct experiment.

For example: In a trial of a gas-engine by Messrs. Brooks and Steward it was necessary, in order to determine the quantity of heat stored and wasted in the exhaust gases, to determine the specific heat of the mixture of steam, carbon dioxide, and nitrogen thus: * The analysis of the gas used in the tests is—

| | | By volume. |
|---------------------------------------------------------------------------|--------------------------------|--------------|
| H | Hydrogen, | .395 |
| CH ₄ | Marsh-gas, | .373 |
| N | Nitrogen, | .082 |
| C ₃ H ₆ , etc. | Heavy hydrocarbons, | .066 |
| CO | Carbonic oxide, | .043 |
| O | Oxygen, | .014 |
| H ₂ O ₁ , CO ₂ , H ₂ S, etc., | Water-vapor, impurities, etc., | .027 |
| | | <u>1.000</u> |

By weight its composition is found to be—

| | Cu. meters. | | Densi- ties.† | | Kilos per cu. m. | W't p. unit. |
|----------------------------------------------|----------------|---|------------------|---|---------------------|-----------------|
| H | .395 | × | .087 | = | .035 | .058 |
| CH ₄ | .373 | × | .694 | = | .258 | .426 |
| N | .082 | × | 1.215 | = | .099 | .163 |
| C ₃ H ₆ , etc. | .066 | × | 1.84 | = | .121 | .200 |
| CO | .043 | × | 1.215 | = | .052 | .086 |
| O | .014 | × | 1.388 | = | .019 | .031 |
| H ₂ O ₁ , etc. | .027 | × | ~.8 | = | .022 | .036 |
| | <u>1.000</u> | × | .606 | = | .606 | <u>1.000</u> |

* Van Nostrand's Magazine, 1883.

† Schöttler: Die Gasmaschine, p. 77.

By "density" is meant the weight of one cubic meter in kilogrammes. One cubic meter of the gas in question weighs 0.606 kilos.

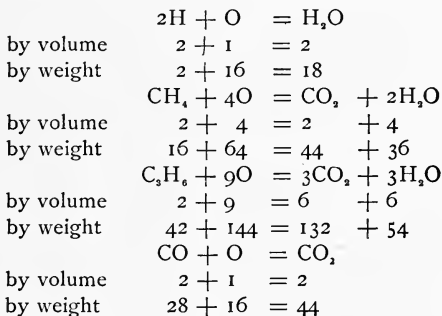
Upon complete combustion the gas develops heat per cubic meter as follows:

| | Calories.* | | Calories. |
|------------------------------------------------|------------|--------|------------------------|
| From H | 29060 | × .035 | = 1020 |
| " CH ₄ | 11710 | × .258 | = 3020 |
| " C ₃ H ₈ , etc. | 11000 | × .121 | = 1330 |
| " CO | 2400 | × .052 | = 125 |
| | | | <u>per cu. m. 5495</u> |

and per kilog. gas $\frac{5495}{.606} = 9070$ calories.

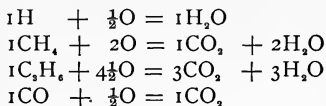
In British measures, one cubic foot of gas develops 617.5 heat-units.

To determine the amount of air supplied for complete combustion, it is necessary to ascertain the quantity of oxygen in chemical combination with the combustible constituents of the gas.



The combining proportions are—

By volume



* Schöttler: Die Gasmaschine, p. 80.

Per unit weight of mixture the composition will be—

| | | |
|------------------|-----------|-------------|
| CO ₂ | | .146 |
| H ₂ O | | .131 |
| N | | .721 |
| Impurities, | | .002 |
| | | <hr/> 1.000 |

The volume which 13.27 kilos of products of combustion will occupy is found from the known volumes of the constituent gases as follows :

| | Kilos. | Cu. m. per kilo. | Cu. m. |
|------------------|--------|------------------------|--------------|
| CO ₂ | . . . | $1.93 \times .524 =$ | 1.011 |
| H ₂ O | . . . | $1.74 \times 1.28 =$ | 2.227 |
| N | . . . | $9.57 \times .823 =$ | 7.876 |
| Impurities, | . . | $.03 \times \sim .9 =$ | .027 |
| | | | <hr/> 11.141 |

The products of combustion occupy 11.141 cu. m. to each kilog. of gas. To find the ratio per cu. meter of gas, we have simply to multiply by 0.606, the number of kilos in a cubic meter, and get 6.751. As 6.94 cu. m. of air and gas are needed to every cu. m. gas, by a contraction of 2.7 per cent. combustion takes place.

The specific heats of the products of combustion are determined from the specific heats of the several component gases as follows :

Specific heat at constant pressure (water = 1) :

$$C_p = \left\{ \begin{array}{ll} .2169 \times .146 \text{ (CO}_2\text{)} & = .0317 \\ .4805 \times .131 \text{ (H}_2\text{O)} & = .0629 \\ .2438 \times .721 \text{ (N)} & = .1758 \\ \sim .4 \times .002 \text{ (impurities)} & = .0008 \end{array} \right\} = 0.2712.$$

Specific heat at constant volume (water = 1) :

$$C_v = \left\{ \begin{array}{ll} .1714 \times .146 \text{ (CO}_2\text{)} & = .0250 \\ .3694 \times .131 \text{ (H}_2\text{O)} & = .0484 \\ .1727 \times .721 \text{ (N)} & = .1245 \\ \sim .3 \times .002 \text{ (impurities)} & = .0006 \end{array} \right\} = 0.1985.$$

The ratio of these specific heats is the exponent of adiabatic expansion, and is found to be

$$\gamma = \frac{C_p}{C_v} = \frac{0.2712}{0.1985} = 1.366.$$

Since there is always an excess of air present, these values will be somewhat modified by that fact. From the meter records the ratio of air to gas by volume was found to be 6.63 to 1; by weight the ratio is

$$\frac{6.63 \times 1.251}{1 \times .606} = 13.68.$$

Since for complete combustion only 12.26 parts of air by weight are needed, there are 1.42 parts in excess. The specific heats of air being $C_p = .2375$ and $C_v = .1684$, the effect of the excess of air will be to reduce the specific heat slightly.

$$C_p = \frac{(.2712 \times 13.26) + (.2375 \times 1.42)}{14.68} = .268;$$

$$C_v = \frac{(.1985 \times 13.26) + (.1684 \times 1.42)}{14.68} = .196;$$

$$\gamma = \frac{C_p}{C_v} = \frac{.268}{.196} = 1.37.$$

87. The Scheme of the Trial should be carefully prepared in advance, and should be so planned as to secure the needed data with certainty and accuracy. The first consideration is the purpose of the proposed trial, and the first work done the arrangement of a general plan that shall enable the observers to collect with accuracy and certainty all the needed data, and to record them conveniently and in most available form. The next matter to be studied is the reduction of all general and special operations of the trial to a complete and efficient system, in which every part shall be made as far as possible contributory to the efficiency and fruitfulness of every

other part; in which each observer shall be so stationed and so instructed that he may secure the data assigned him for collection with least difficulty, risk, and uncertainty, and shall have his own work checked, and shall aid in checking the work of others, as completely as possible. No essential data should remain unchecked, and every subsequent calculation based upon them should also be made by at least two computers independently.

The plan of the work being settled upon, each detail should be studied by itself, and every provision that experience and foresight can suggest should be taken to insure perfection of the scheme. A preliminary and informal trial will then be likely to reveal any serious defect, which being corrected, the final and official trial may be fully expected to give thoroughly reliable results.

88. Competitive Trials of Engines are sometimes conducted by the engineer, either to determine which of two or more competing forms of engine is to be accepted by the purchaser, or by his client, or, as at exhibitions of various kinds, simply to ascertain the power and efficiency of two or more engines, with a view to deciding their relative merits as types of engine, or as representing the best practice of their builders. It is largely through this kind of competition that the best known systems of standard engine-trial have been developed. Examples of such are illustrated by the following regulations, adopted at the exhibitions of the Franklin Institute of the State of Pennsylvania:

NOTICE:—Exhibitors of engines, desiring quantitative tests made of their exhibits, must make formal application for such tests in advance.

Engines can be exhibited, but will not be tested unless formal application and agreement to the following code are completed within the specified time.

Parties desiring tests made of their engines can have them made by making formal application therefor, and by subscribing to and fulfilling the conditions of the Code.

All tests will be quantitative, and will, once begun, not be abridged, save by special agreement with the judges.

Tests of regularity of speed, however, will be made independently of other measurements.

The Committee reserves the right to limit the number of engines tested and to elect which engines shall be tested, if time will not permit complete tests for all making formal application.

Competitive tests will not be made, save on the joint application of the two or more parties desiring them, who must, previous to the tests, agree on the rating of the various points of the engine (see Article 9), and subscribe to the Code, agreeing to abide by the decision of the judges without appeal.

Conditions of Exhibition and Test.

(1) The cylinders of the engines entered may be of any capacity and proportion of stroke to diameter.

(2) Each cylinder shall be drilled and tapped by the builder, for indicator connections, by means of one-half ($\frac{1}{2}$) inch pipe in the usual manner, and to the satisfaction of the judges. Pet drainage-cocks must be on the cylinder. The cross-head or some other point must be drilled for the indicator-cord attachment.

(3) Each cylinder shall be drilled and plugged at both ends so as to admit of being completely filled with water and emptied by means of a one-half ($\frac{1}{2}$) inch pipe, in order to determine the clearance and the piston-displacement of one stroke at each end. These data will be obtained both hot and cold.

(4) The steam and exhaust valves will be tested under full steam-pressure, ninety (90) pounds per square inch by the gauge, unless some other pressure has been agreed upon for the test.

(5) The tightness of the piston-packing will be determined by removing the back cylinder-head and subjecting the piston to full boiler-pressure on each centre.

(6) Each maker is requested to use such diameter of band

fly-wheel or of pulley as shall give a belt-speed of 4000 feet per minute. Should he require a different belt-speed, he will specially note the same, in communicating with the Committee.

(7) Each exhibitor will be required to furnish his own connections with the main steam-pipe, the main injection-pipe, and the main overflow pipe or tanks.

(8) Each exhibitor will be furnished with space at the regular rates established for the exhibition, in which space he must build his foundations at his own cost, and subject to the approval of the Superintendent.

(9) Each exhibitor will communicate to the chairman of the Committee such a description and drawings of the engine exhibited as will facilitate the labors of that Committee, together with his claims as to meritorious points for his exhibit.

The following points will have special consideration : *

- | | |
|--------------------------------|-------------------------------|
| 1. Economy of steam. | 5. Simplicity of design. |
| 2. Regularity of speed. | 6. Perfection of proportions. |
| 3. Concentration of power. | 7. Finish of parts. |
| 4. Durability of construction. | |

Each exhibitor must file the following data, before the tests, viz. :

Diameter of steam-cylinder to nearest hundredth of an inch.

Diameter of piston-rod " " "

Diameter of steam-pipe " " "

Diameter of exhaust-pipe " " "

Diameter of fly-wheel " " "

Width of the face of fly-wheel " " "

Weight of fly-wheel in pounds " " "

Area of steam-ports, each to nearest hundredth of an inch.

Area of exhaust-ports, " " "

Stroke of engine, " " "

Indicated horse-power of engine when believed to be working most economically.

Revolutions of crank per minute.

Weight of whole engine, exclusive only of fly-wheel.

* These are the points referred to in the special notice concerning the value of which agreement must be had previous to the competitive tests.

When a condenser is used and its air-pump driven by the engine, the following additional data will be required, viz.:

Diameter of air-pumps to nearest one-hundredth of an inch,

Diameter of injection-pipe " " "

Diameter of overflow-pipe " " "

Stroke of air-pump piston " " "

And if an independent condenser is used, *i.e.*, not driven by the engine, give

Diameter of injection-pipe to nearest one-hundredth of an inch,

Diameter of overflow-pipe " " " "

Drawings of condenser used, any other data peculiar to it, and a full description.

Preparations for the Tests.

(10) The steam for the tests will be furnished by the exhibition-boilers, and will come from boilers specially set apart for the purpose of the tests. It will be charged for at regular rates of three (3) cents per indicated horse-power per hour. Steam will be furnished to exhibitors one week before the tests are made, if desired.

No charge will be made for the services of attendants or experts, or the use of apparatus, unless in some extraordinary case, when the cost will be fixed by the superintendent.

(11) The steam-pressure used will be subject to the wish of the exhibitor, but shall not exceed ninety (90) pounds per square inch, by the gauge.

A special standard gauge will be used during the tests, and subjected to careful tests before and after use.

(12) The safety-valve will be set to blow off at ten (10) pounds above the pressure fixed upon.

(13) The thermal value, the temperature, and the pressure will be taken by means of scale-calorimeters, thermometers, and standard gauges at the boiler, at the steam-chest, and at the exhaust, if the engine is non-condensing.

The thermometers, calorimeters, etc., will be furnished by the exhibition, but the exhibitor must do such mechanical work, must furnish such piping, tools and materials, as are neces-

sary to make the required attachments, at his own cost, and subject to the orders of the Committee.

(14) The temperatures of injection and of hot-well will be taken with standard thermometers, in the case of condensing-engines.

(15) The water used will be taken from the city mains.

The feed-water for the boilers will be weighed by means of scales and a large tank, and will be run into a smaller supplemental tank, from which it will be pumped into the test-boilers by means of a feed-pump actuated by steam from other boilers.

The condensing water used will, in the case of condensing-engines, be measured after leaving the hot-well, in two carefully gauged tanks, alternately filled and emptied, the temperature also being taken.

The known weight of steam used will be subtracted from the overflow.

The injection-water will be weighed in large tanks, and its temperature taken.

The injection-water will not be delivered under pressure.

(16) The number of revolutions of the engines will be taken by a continuous counter attached to the crank-shaft.

The variations in speed for one minute will be taken at each quarter of an hour by means of an electric chronograph, connected with a standard clock, beating seconds.

The variations in speed during one stroke will be taken by an acoustic chronograph at fifteen minutes' intervals.

Special tests of speed alone, under varying loads, will be made if desired, and close attention will be had to this point in all cases.

(17) A standard barometer and thermometer will be read at fifteen-minute intervals during the trial.

(18) The vacuum of condensing-engines will be read by a gauge, carefully compared before and after the trials.

(19) All of the gauges, indicators, and thermometers used shall be carefully tested before and after the trials, and the

party whose engine is tested shall have the right to be present in person or by agent at these tests.

(20) The indicator-diagrams will be taken at fifteen (15) minute intervals, and will be read for—

- Initial pressure,
- Pressure at cut-off,
- Terminal pressure,
- Counter-pressure at mid-stroke,
- Maximum compression-pressure,
- Mean effective pressure,
- Point of cut-off,
- Release of steam,
- Exhaust closure.

From the diagrams will be computed the indicated steam at the point of cut-off and at release, as also the actual steam from boilers per horse-power per hour.

(21) The Committee will test the engine at the load desired by the exhibitor of it, unless circumstances shall render it impossible to meet his wishes.

If the load does not exceed seventy-five (75) indicated horse-power, the net load will be measured by a transmitting dynamometer.

(22) At the close of the regular trial the engine will have its belt taken off, and be run for one hour for friction-diagrams.

(23) Unless otherwise arranged, the trials will last ten (10) hours.

(24) No account will be taken of the coal burned, but the economy of the engine will be deduced from the actual steam used and water weighed to the boiler.

The trial will begin with the established pressure.

The level of the water in the boiler and the pressure of the steam will be kept as nearly constant as possible during the whole of the trial.

The whole weight of the water fed to the boiler, subject to proper deductions for waste, and to corrections for variation of level in the boiler, will be multiplied by its thermal value as steam at the steam-chest, and divided by the product of the

indicated horse-power of the engine, and the number of hours of the test.

The resulting quotient will be used to divide twenty-five hundred and fifty-seven and sixty-nine one-hundredths (2557.69) British thermal units, * giving the efficiency of the engine as compared with the mechanical equivalent of the heat furnished to it, and therefore its efficiency, as a means of converting heat into work.

The net horse-power of the engine will be used for computation similarly to the indicated horse-power, and the result will be taken as the measure of the efficiency of the engine, both as a means of converting heat into work, and as a machine for the transmission of power.

This latter shall be considered the true measure of the efficiency of the engine.

89. Regulations for Competitive Boiler-trials are illustrated by the following, adopted at the same time as the preceding :

NOTICE.—Boilers may be exhibited and used at the Exhibition, but quantitative tests of their efficiency will not be made except upon formal application, and the acceptance of the subjoined code.

Competitive tests will not be made unless at the joint request of the parties desiring them and until such parties have agreed to and subscribed to this code, and fixed upon a rating for the points enumerated in Article 4.

The Committee reserve the right to limit the number of tests made, should time and opportunity not permit the completion of all the tests desired.

Preliminaries to the Tests.

(1) *Capacity*.—The boilers entered may be of any capacity having an evaporative power not less than seven hundred and fifty (750) pounds of water per hour.

Each boiler must be so drilled and fitted with proper pipes and cocks that the judges may be enabled to determine readily

* Joule's Equivalent is here taken as 774.1 foot-pounds.

its whole water capacity by filling and emptying the boiler and weighing the contents.

(2) *Pipes and Valves*.—Each exhibitor will furnish all the pipes and valves necessary to make connection with the main water and steam pipes in a proper manner, and subject to the orders of the Superintendent.

He will also make any alterations in water and steam pipes required for the tests, furnishing all tools, piping, cocks, and mechanical labor at his own cost.

(3) *Space*.—Each exhibitor will be furnished with space at the regular rates established for the Exhibition, in which space he must build his foundations and boiler-setting, and make connection with the chimney-flue, if required, at his own cost, and subject to the approval of the Superintendent.

(4) *Specifications*.—Each exhibitor must furnish to the Chairman of the Committee such description and drawing both of the boiler in position and of the details of the boiler as will facilitate the labor of that Committee, together with his claims as to the meritorious points of his exhibit.

The following points will have special consideration :

1. Economy of fuel; 2. Economy of material and labor of construction; 3. Evaporative power; space occupied; 4. Simplicity and accessibility of parts; 5. Durability of whole structure.

Exhibitors desiring a competitive test made must agree upon a rating for these points before it will be made.

Exhibitors must also file the following data :

Area of heating-surface to the nearest hundredth of a foot ; area of grate-surface to the nearest hundredth of a foot ; area of calorimeter-surface to the nearest hundredth of a foot ; area of chimney-flue surface to the nearest hundredth of a foot ; height of chimney desired ; number of pounds of coal per square foot of grate to be burned per hour.

Should the determinations of these preliminaries by the Committee differ in result from those of the exhibitor, he will be required to give all the details of his calculations, and an agreement must be reached before proceeding with the test.

Preparations for the Tests.

(5) *Coal*.—Anthracite coal will be used and will be furnished free of charge, provided the steam made is used for the general purposes of the Exhibition.

The same quality and size of coal will be used in all the tests, unless special arrangements be made for another kind of fuel.

An analysis will be made of the coal used.

The coal will be weighed to the boiler.

(6) *Water*.—The water used will be taken from the city mains.

The feed-water for the boilers will be weighed by means of scales and a large tank, and will be run into a smaller supplemental tank, from which it will be pumped into the boiler by means of a feed-pump actuated by steam from the boilers.

The temperature of the feed-water will be taken by means of a standard thermometer in the supplemental tank.

(7) *Pressure*.—The steam-pressure used shall not exceed ninety (90) pounds per square inch by the gauge, unless by special arrangement with the Committee.

A standard gauge will be used, and also a standard thermometer immersed in a mercury-pocket in the steam-space.

(8) *Safety-valve*.—The safety-valve will be set to blow off at ten (10) pounds above the pressure fixed upon.

(9) *Leaks*.—Within twenty-four (24) hours preceding the test of the boiler it must be subjected to hydraulic pressure ten (10) pounds greater than its steam-pressure during the test, and proved to be perfectly tight.

(10) *Attendants*.—The attendants in charge of the boiler tested must be approved by the party whose boiler is tested and by the judges. All attendants are to be subject to the orders of the judges during the progress of the test.

(11) *Ashes*.—All ashes will be weighed on being withdrawn from the ash-pit, and must not be damped until weighed.

(12) *Calorimeters*.—The calorimeters used will consist of a barrel, scale, and hand thermometer.

Two calorimeters will be used and simultaneous observations made at fifteen (15) minute intervals.

(13) *Fires*.—The exhibitor shall be allowed one day previous to the test to clean boilers and grates.

The steam having reached the required pressure, the ash-pit shall be thoroughly cleaned and swept, and thereafter the fire maintained as nearly uniform as possible, the test closing with the same depth and intensity of fire as it opened.

This point is to be decided by the judges, who may make allowances if it be clearly shown to have been impossible to maintain uniform fires.

If in the judgment of the Committee the firing is inefficiently or improperly done, the test may be terminated at any time, and a repetition of the test granted or refused.

(14) *Pyrometer*.—The temperature of the gases of combustion immediately upon entering the chimney-flue shall be taken by means of a suitable pyrometer, read at fifteen (15) minute intervals, and close to the boiler.

(15) *Manometer, Barometer*.—The vacuum in the chimney-flue shall be taken by means of a water-manometer, read at fifteen (15) minute intervals, if natural draught is used.

If a forced blast is used the manometer will be placed on the conduit to the ash-pit.

A barometer will be read simultaneously.

(16) *Duration*.—Unless otherwise arranged, the tests will last ten (10) hours.

(17) *Economy and Efficiency of the Boiler*.—The level of the water in the boiler and the state of the fire must be kept as nearly constant as possible during the whole of the trial.

The weight of the water in the boiler for each one-quarter of an inch, on the glass water-gauge, will be carefully determined and recorded previous to the test, and proper correction for unavoidable changes of level made.

The weight of water fed to the boiler, subject to proper corrections, will be multiplied by its observed thermal value as steam.

From this product thermal units of heat brought in by the feed will be subtracted.

The remainder will be divided by nine hundred and sixty-five and seven-tenths (965.7) British thermal units,* giving the number of pounds of water evaporated from and at 212 degrees Fahrenheit.

This latter quantity will be divided by the weight of coal burned, less weight of dry ashes, giving the number of pounds of water evaporated per pound of combustible.

This shall be taken as the measure of the efficiency of the boiler.

(18) *Nominal Horse-power*.—The nominal horse-power of the boiler will be deduced by dividing the number of pounds of water evaporated from and at 212 degrees Fahrenheit per hour by 30.

(19) *Evaporative Power*.—The evaporative power of the boiler will be determined by dividing the nominal horse-power of the boiler by the number of cubic feet of space it occupies.

The space occupied by a boiler and its appurtenances will be regarded as the product of the square feet of floor-space occupied by its extreme height in feet.

Steam-pump tests have been conducted at such exhibitions under the following regulations, written, originally, by Mr. Hill:

Regulations for Test of Steam Pumps.

(1) Steam will be furnished by the boilers used in the experiments upon automatic and slide-valve engines; the pressure will be taken in the pipe as near the stop-valve as convenient. The pressure in the boilers will be maintained as uniformly as possible at (75) seventy-five pounds per sq. inch above atmosphere.

(2) A calorimeter test of the quality of steam furnished will be made regularly every thirty (30) minutes. The steam-pipe will be tapped in the last horizontal joint toward the pumps for calorimeter connection.

* The value taken here for the latent heat of steam at the boiling-point.

(3) The exhaust will be delivered to a surface-condenser having not less than 500 sq. feet of condensing surface; the condensing water will be obtained from the city mains; water of condensation will be collected in a tank placed under the outlet nozzle of condenser.

(4) The suction tank will be placed below the level of pump; the distance from bottom of tank to centre of water-cylinder will be uniform. For all contestants the vertical head of water in suction-tank will be taken with a sliding-hook gauge, at regular intervals.

(5) The delivery-tank will be placed on a staging directly over the water-cylinder of pump, the discharge opening of water-cylinder will be connected with a 6-inch vertical stand-pipe furnished with a direct-weighted safety-valve; the height of stand-pipe from centre of water-cylinder to centre of orifice of safety-valve will be 10 feet. The safety-valve will be loaded to create a resistance per sq. inch equivalent to a dynamic head of 150 feet less the height of stand-pipe (10) feet.

(6) The measuring-tank will be placed (vertically) between section-tank and delivery-tank; the measuring-tank will be divided by a vertical partition in the centre into two compartments. Each compartment will have a capacity of 300 cubic feet; the water will be delivered from the receiving-tank into the measuring-tank through a (6) six-inch swinging nozzle. The nozzle will be directed over one tank until it has been filled and the water breaks over the dividing partition, when it will be swung over the empty tank; in the mean time the temperature of the water in the full tank will be noted, the number of tank entered in the log, and the contents drawn off through an (8) eight-inch delivery-pipe into the suction-tank below; this operation will be repeated regularly during the run. The precise capacity of each compartment of the receiving-tank will be determined prior to the experiments by filling each to the crest of partition, and drawing off, weighing, and noting the temperature of contents.

(7) The duration of run will be fixed at (5) five hours. Previous to the commencement of run, the steam will be turned

on and the pump will be operated until all parts have acquired the working temperature.

(8) The pressure of atmosphere will be taken from a United States standard mercurial barometer.

(9) Thermometers will be located as follows: No. 1 in barometer case to note the temperature of atmosphere. Nos. 2 and 3 in the two compartments of measuring-tank. No. 4 in calorimeter.

(10) The time of commencement and close of run and periods of observation will be taken from a chronometer clock placed near the pump under experiment; the periods of observation will be indicated by a double stroke of the signal-gong; one minute previous to each observation a single stroke of the gong will be made calling the assistants to their stations. Every (15) fifteen minutes a full set of observations will be made and entered in the log.

(11) A revolution counter will be connected to standard on piston-rod.

(12) Previous to experiments, each exhibitor will hand to the Board of Experts a complete schedule of dimensions of steam and water cylinder, internal diameter of steam and exhaust pipes, area of steam-ports, internal diameter of suction and delivery-pipes and volume of clearance in steam-cylinder.

(13) The economy will be determined by the water of condensation collected in the tank under condenser, corrected by the average of result obtained from calorimeter observations; and the cost of the work in coal (Pittsburgh No. 1) at $\frac{1}{3}$ this upon assumed boiler efficiency of (9) nine pounds water evaporated per pound of coal burned on the grates.

(14) The duty will be stated in gallons lifted one foot high.

90. Standard Systems of Boiler-trial have been already discussed and described at such length that it is unnecessary to add anything more here than to remark that, in every case of real importance, the careful and skilful management of the boiler-trial as a part of the whole work, in the measurement of the efficiency of the system of heat-production and utilization, becomes an essential element of success. The best standard

methods, as a whole and in every detail, should be adopted. Unless the measurement of the quantity and quality of the steam supplied be accurately made, it is quite impossible to obtain a correct measure of the efficiency of the apparatus in which it is utilized by conversion into work and power.

91. The Heat-energy, the Quantity and Quality of Steam used, and the availability of the heat stored in that steam and transferred to the engine with it, can only be exactly known when the weight, the wetness or dryness, the pressure, and the thermal properties of the steam are precisely ascertained. The weight of feed-water pumped into the boiler is the weight of the mixture of steam and water, if any water is entrained by the steam, which is sent to the engine. The heat so supplied, diminished by the usually simple waste from the exterior of the steam-pipe, is the amount received by that machine. The availability of that heat for its purpose depends upon the degree in which the temperature and the pressure of the steam exceed the temperature and pressure of the atmosphere or of the condenser.

The first point to be attended to is the testing of the steam to ascertain whether it be wet, dry and saturated, or superheated; and if not dry and saturated, to what extent it stores an excess per unit of weight by superheating, or a deficiency of heat in consequence of its admixture with water. This is determined by the use of the calorimeter.

Smoke-preventing apparatus is sometimes attached to boilers, and it becomes important to determine the quality of the products of combustion in this respect. This usually involves a boiler-trial and a comparison with ordinary furnaces. It will often be found that the prevention of smoke involves an excessive air-supply and a consequent waste of fuel and loss of efficiency. This may be partly compensated, however, by the improved performance of the boiler, due to cleaner and more effective heating-surfaces and the absence of soot deposits.

92. Special Methods of Engine-test are sometimes adopted in competitive trials of special forms, as illustrated by the following regulations prepared by Messrs. Hill and Holmes.

Code of Regulations for Tests of Automatic Cut-off Engines.

(1) Steam will be obtained from a pair of locomotive fire-box boilers, furnished by the Commissioners. These boilers have a combined evaporative capacity of 2250 pounds of water per hour; a heating-surface (combined) of 983 square feet; and a grate-surface of 12.7 square feet. Each boiler will be provided with a safety-valve, loaded to blow off at 85 pounds pressure above the atmosphere. Each boiler will have attached, independently, an accurate test-gauge, and if it can be procured, an Edson recording-gauge. The height of the water will be indicated by a glass water-gauge on each boiler, in addition to the usual test-cocks.

(2) The feed-water will be weighed in the receiving-tank, and drawn off as occasion requires into the supplemental tank. The water will be supplied to the boilers by an independent steam-pump, having the suction connected with the supplemental tank, and the discharge with check-valves of boilers. The steam to drive the boiler-feeders will be obtained from boilers independent of those furnishing steam for the engine under experiment. The water fed into the boilers will be determined in weight whilst in the receiving-tank. The receiving-tank will have a capacity of 2300 pounds water, at 175° Fahr.; the supplemental tank will have a capacity of 1000 pounds water at 150° Fahr.

(3) The resistance will be obtained by a 100 horse-power blower, having a sliding iron gate fitted to its discharge orifice. The position of the gate having been determined, it will be fastened at this point during the experiment.

(4) A pair of indicators will be attached to the cylinder of engine, one at each end. The indicators will be moved in such a manner that the diagrams shall be coincident with the motion of the piston.

(5) Two engine-counters will be employed, one to indicate the revolutions of the main shaft of engine, and one to determine the revolutions of the jack-shaft.

(6) The dynamometer (transmitting) will be keyed to jack-shaft, between the pulley receiving the belt from the engine, and the pulley carrying the blower belt.

(7) A steam-gauge will be screwed into the steam-pipe as near the stop-valve as convenient.

(8) The pressure in the chest will be determined by a chemical thermometer immersed in a cup of mercury, screwed into the steam-space of the chest. A test-gauge will also be screwed into the steam-chest to determine the effect of part closure.

(9) The temperature of the feed-water will be taken on a mercurial thermometer located in the supplemental tank.

(10) The temperature of the cylinder clothing will be taken on a thermometer with the bulb in contact with the outer covering of the cylinder.

(11) The pressure of the atmosphere will be taken with U. S. standard mercurial barometer; the temperature of atmosphere will be read on the thermometer in barometer case.

(12) The time will be determined by a chronometer clock placed near the engine under experiment.

(13) The time of noting observations will be indicated by a double stroke of the signal-gong; one minute previous to each observation a single stroke of the gong will be made, calling the assistants to their stations.

(14) Previous to experiments, all pipe connections with the boilers will be carefully closed, leaving open only the steam-pipe connecting with engine, and feed-pipe connecting with the pumps.

(15) Each exhibitor will hand to the Board of Experts, previous to the experiments, a complete summary of the dimensions of his engine, including the volume of clearance, steam and exhaust port area, and weight of reciprocating parts.

(16) The duration of experiments will be fixed at eight hours. Previous to the beginning of experiments, the boilers will be steamed up to the running pressure, and the height of water brought to the thread tied around the middle of the glass tubes in the water-gauges. All water supplied to the boilers

thereafter will be weighed and charged to the engine. The height of water at close of experiment will be made to coincide with the thread on the glass tubes.

(17) Every fifteen minutes a full set of observations will be made and entered in the log.

(18) During the economy test, the engine will be run with full opening of stop-valve.

(19) The economy of the engine will be determined upon the consumption of water per I. H. P. per hour, and the cost of the power (in coal) one-ninth this upon an assumed evaporative efficiency of boilers of nine pounds of water per pound of coal.

Code of Regulations for Tests of Slide-valve Engines.

(1) Steam will be supplied from the boilers used in the test of automatic engines. The general dimensions are stated in paragraph 1 of the Code of Regulations for Experiments upon Cut-off Engines.

(2) The economy will be determined upon the consumption of water per I. H. P. per hour. The water will be delivered from the exhaust-heater into the receiving-tank, where it will be weighed and entered in the log of the engine under experiment. The water will be drawn from the receiving-tank into the supplemental tank connected with the suction of the pumps feeding the boilers. The steam to drive the boiler-feeders will be obtained from boilers independent of those furnishing steam for the engine under experiment. The receiving and supplemental tanks will be the same as used in the cut-off experiments. The dimensions are enumerated in paragraph 2 of the Regulations for the Test of Cut-off Engines.

(3) A calorimeter-test of the quality of steam furnished will be made regularly every 30 minutes.

(4) The power will be absorbed by a 100 H. P. pressure blower. This will have an adjustable gate fitted to the discharge-orifice to regulate the resistance. The area of opening will be fixed during the run.

(5) Diagrams from each end of cylinder will be taken. The

motion of indicator-drum will be such as to produce a diagram coincident with the movement of piston.

(6) The counter indicating the revolutions of engine will be connected direct. The counter showing revolutions of jack-shaft (carrying dynamometer) will be driven by positive connectors at a reduced speed.

(7) The dynamometer (transmitting) will be keyed to jack-shaft between the pulley receiving the belt from the engine, and the pulley carrying the blower-belt. The indications of the dynamometer will be read from a station in close proximity to the instrument.

(8) A test-gauge of approved make will be screwed into the steam-pipe as near the stop-valve as convenient. The initial pressure in the cylinder will be compared with the pressure in the pipe.

(9) Thermometers will be used as follows: No. 1 to show the temperature of atmosphere; 2, in the feed-water tank; 3, in the steam-chest of cylinder; 4, in the calorimeter. The thermometers to be U. S. standard instruments, thoroughly tested before the experiments, and of uniform scale. No. 1 will indicate temperatures from 32 to 120 degrees Fahr.; Nos. 2 and 4, from 32 to 250 degrees; and No. 3, from 32 to 600 degrees Fahr.

(10) The pressure of atmosphere will be read from a U. S. standard compensated aneroid barometer.

(11) The time of commencement and close of run, and intervals of observations for the log will be taken from a chronometer clock, placed near the engine under experiment. The time of noting observations will be indicated by a double stroke of the signal-gong. One minute previous to each observation a single stroke of the gong will be made calling the assistants to their stations.

(12) Previous to the experiments all pipe connections with boilers will be carefully closed, leaving open only the steam-pipe connecting with the engine and feed-pipe connecting with the pumps. Great care will be taken that all steam generated in the boilers be delivered to the engine.

(13) The duration of run will be fixed at five (5) hours. Previous to the beginning of experiment, the boilers will be steamed up to the running-pressure (75 lbs. above atmosphere), and the height of water brought to the thread around the tube in glass water-gauge. All water delivered to the boilers thereafter will be regularly entered, by weight, in the log. The condition of pressure and height of water will be maintained as nearly uniform as possible during the run, and made to coincide with the initial conditions at close of run.

(14) During the economy-test the engine will be operated with an open stop-valve.

(15) At close of economy-run the main belt will be thrown off, and the engine throttled to run at load speed for the friction-diagrams.

(16) Previous to the experiments each exhibitor will hand to the Board of Experts a complete summary of the dimensions of his engine, including volume of clearance, steam, and exhaust-port area (least), weight of reciprocating parts, and estimated I. H. P., at 75 pounds pressure in the pipe.

(17) Every fifteen minutes a full set of observations will be made and entered in the log.

(18) The cost of the power in coal will be taken at one ninth the consumption of water per I. H. P. per hour upon an assumed boiler efficiency of nine pounds water evaporated per pound of coal burned on the grate.

Code of Regulations for Tests of Mounted (Portable) Engines.

(1) Steam will be furnished each engine by its attached boiler. Each exhibitor will be required to hand to the Board of Experts, previous to the experiments, a schedule of the length, width, and height from grates to crown of fire-box; total heating-surface; number, length, and external diameter of tubes; thickness of water-leg; diameter and vertical and horizontal distance apart of stay-bolts, diameter and length of barrel, total area of openings through fire-bars, area of openings through ash-pit door, thickness of iron in shell and fire-box,

cubic feet of water carried to the gauge-line, cubic feet of steam room.

(2) Before the commencement of experiments the boiler will be steamed up to the running-pressure, using for fuel. Each exhibitor will have weighed to him a sufficient quantity for the economy-run. The management of the fire and the use of the fuel will be entirely under the control of the exhibitor. All fuel remaining in the pile at close of run, and all unburnt wood on the grates, will be weighed back to the credit of the exhibitor. The ashes under the grate will be weighed back dry.

(3) A calorimeter-test of the quality of steam furnished by the boiler will be made regularly every 30 minutes.

(4) The dynamometer will be keyed to main shaft of engine; the power will be taken off from a pulley attached to loose side of dynamometer; the resistance will be created by a 40 H. P. pressure blower, the discharge-orifice of which will be fitted with an adjustable gate; the area of discharge-opening will be fixed constant during the run.

(5) One revolution-counter will be employed, having a positive connection with the valve-stem of engine.

(6) The feed-water will be weighed in the receiving-tank, and passed thence to the supplemental tank, from which it will be drawn by the boiler-feeder. The water will be delivered from the city mains (unheated) into the receiving-tank. Each engine will heat its own feed-water.

(7) The duration of run will be fixed at five (5) hours. Previous to the experiment, the engine will be run without load until all parts have acquired the working-temperature and the water brought to the thread tied around the tube of glass water-gauge. All water fed to the boiler from commencement to close of run will be regularly weighed and entered in the log. The pressure of steam and height of water at close of experiment will be made to coincide with the initial conditions.

(8) Thermometers will be located as follows: No. 1, in

barometer-case, to indicate the temperature of atmosphere; No. 2, in feed-water tank; No. 3, in calorimeter.

(9) The pressure of the atmosphere will be taken from a U. S. standard compensated aneroid barometer.

(10) The pressure of steam in the boiler will be read from a reliable steam-gauge, independent of that belonging to the engine.

(11) Previous to experiment, all pipe connections with boiler or engine will be detached, except the pipe from steam-dome to cylinder, and the suction-pipe from feed apparatus to supplemental tank.

(12) The economy will be determined upon the consumption of coal per I. H. P. per hour.

(13) Before the experiments begin, each exhibitor will hand to the Board of Experts a complete summary of the dimensions of engine, including volume of clearance, area of steam and exhaust ports, internal cross-section of steam-pipe, and weight of reciprocating parts. The valve-motion will be shown by a single indicator-diagram from each end of the cylinder, taken at a uniform piston-speed.

(14) During the economy-run the engine will be operated with an open stop-valve.

(15) At close of economy-run the main belt will be thrown off, and the engine throttled to run at load speed. Friction-diagrams will then be taken from each end of cylinder.

(16) Every fifteen minutes a full set of observations will be made and entered in the log.

(17) The time of commencement and close of run, and period of observation, will be determined by a chronometer clock placed near the engine under experiment. The time of noting observation will be indicated by a double stroke of the gong. One minute previous to each observation a single stroke of the gong will be made, calling the assistants to their stations.

CHAPTER IX.

EXAMPLES OF ENGINE-TRIALS.

93. Examples of Engine-testing, as illustrating current and standard practice, will better complete a treatise on this subject than any further extended descriptions of details and of methods of observation, of computation, and of preparation of reports. In the following pages are given, as fully as is possible without occupying too great space, illustrations of this character, as obtained by reference to the reports of the more expert and most experienced of contemporary engineers, or to reports of earlier work which have been regarded by engineers as best representing good practice in special departments.

In all cases the engineer must himself judge whether to err, if at all, in making such tests and in preparing his report, in the direction of extended and complete—and hence costly—investigation and deduction, or in that of brevity and possible incompleteness. The general rule should be to secure all data essential to the purposes of the trial, and, incidentally, to secure all additional facts and data that he may find attainable without incurring objectionable expense. If, in any case, a doubt arises, it will usually be wisest to err on the side of completeness and accuracy. Thus, when called upon, as an expert, to ascertain whether the terms of a contract specifying simply the duty of a pumping-engine have been fully complied with, he need only measure, usually, the coal consumption, the quantity of water pumped by the machine, and the head against which the water is raised. To attempt more would sometimes be unnecessary, and might involve considerable unauthorized expense. But the expert engineer may often, without appreciable additional cost, obtain valuable data relating to the distribution, the utilization, and the wastes of heat, of

energy, and of steam and fuel: he should in such case endeavor to obtain all such data and make them useful.

The following selected illustrations cover the ground very completely, and will probably be quite sufficient for their purpose.

94. Illustrations of the Trial of Stationary Engines abound in the current periodicals, and may be referred to by the engineer seeking to make comparisons. As a model of brevity in reporting such a trial the following is selected. It is the report of Mr. Flower on a trial of a small Corliss engine designed by Mr. Edwin Reynolds.

"GENTLEMEN: I have made an economy test of your engine and boilers as directed by you, and beg leave to report thereon as follows:

| | | |
|---------------------------------|--------------------------------------------------------|----------------|
| Date. | Test of engine made..... | Aug. 3d, 1882. |
| Description of Engine. | Reynolds' Corliss..... | |
| | Diameter of cylinder..... | 14 in. |
| | Length of stroke..... | 36 " |
| | Clearance (assumed)..... | .025 |
| | Nominal horse-power at 82 Rev..... | 68 |
| Trial. | Trial began at..... | 7.30 A.M. |
| | " ended "..... | 4.06 P.M. |
| | Duration of trial..... | 8 h. 36 m. |
| Revolutions. | Revolutions of engine during test..... | 35,372 |
| | " " per minute..... | 69.22 |
| | Temperature of engine-room..... | 99° |
| Gauge-pressure. | Maximum pressure in boiler..... | 72 pounds. |
| | Minimum " "..... | 52 " |
| | Average " "..... | 65 " |
| | Variable boiler-pressure..... | 80 " |
| Pressure per Indicator. | Average initial pressure on piston..... | 61.2 " |
| | " total " "..... | 27.08 " |
| | " gross effective pressure on piston..... | 25.58 " |
| | Per cent. of boiler-pressure appearing as initial..... | 94 |
| Counter-pressure per Indicator. | Back-pressure due to friction through ports..... | .45 pounds. |
| | " " " cushion..... | 1.05 " |
| | " " total on piston..... | 1.50 " |
| Indicated Horse-power. | Maximum gross effective horse-power..... | 66.00 |
| | Minimum " " " "..... | 41.40 |
| | Average " " " "..... | 49.30 |
| | Average total " " " "..... | 52.20 |

| | | |
|------------------|---------------------------------------------------------------------------------------------|------------------|
| Friction | Maximum friction horse-power..... | 27.21 |
| Diagrams. | Minimum " " " | 22.88 |
| | Average " " " | 24.32 |
| Distribution | Average total horse-power..... | 52.20 |
| of Load. | " gross effective horse-power..... | 49.30 |
| | " net " " " | 24.98 |
| Dynamo, Horse- | Edison's dynamo required..... | 4.58 h. p. |
| power of. | Brush " " | 11.54 " |
| Water used, | Temperature of feed-water..... | 203° |
| actual. | Total water pumped into boiler..... | 11,520.5 pounds. |
| | " " " " " per hour.... | 1,340.0 " |
| | Moisture in steam..... | 3 per cent. |
| Steam per hour, | Dry saturated steam per hour..... | 1300 pounds. |
| actual. | Dry steam per indicated horse-power, per hour..... | 24.9 " |
| | Dry steam per gross effective horse-power per hour..... | 26.36 " |
| Steam used, as | Steam account edfor by indicator per hour | |
| shown by In- | per horse-power, total..... | 24.37 " |
| dicator. | Efficiency of cylinder..... | .97 |
| Coal per hour | Coal per hour per horse-power, anthracite... | 3.03 |
| per H. P. | Combustible " " " | 2.60 |
| Cost. | Cost per day of 10 hours, coal @ \$6.25 ton. | \$4.94 |
| Coal. | Coal " " anthracite | 158 pounds. |
| Indiana block | Coal per hour per horse-power..... | 3.62 " |
| coal. | " per day of 10 hours..... | 1,890 " |
| | Cost " " @ \$3.75 ton..... | \$3.54 |
| Maximum Dia- | Indicated power of heaviest diagram..... | 66 h. p. |
| gram No. 31. | Cut-off in parts of stroke..... | 3.60 |
| | " from commencement of stroke..... | 10 inches. |
| | Initial pressure..... | 64 pounds. |
| Possibilities of | Same initial pressure as above..... | |
| Engine. | Diagram, $\frac{1}{2}$ cut-off, engine will develop.... | 72 h. p. |
| | Same cut-off as No. 31, and 74 lb. initial pressure..... | 70 h. p. |
| | Same cut-off and same initial pressure as No. 31, with 82 rev. per minute will develop.. | 77 " |

" When No. 31 was taken, all the machinery that could be put on was on. The average power used was much less than No. 31, being 49.3 horse-power.

"As is shown in body of report, the power of the engine may be increased by increasing the revolutions to 82 per minute (that being the rate of speed as given by the builder), by allowing a later cut-off, or by a higher initial pressure.

"A higher initial pressure may be had by increasing the boiler-pressure ; the average during test was 65 pounds, and you are allowed 80 pounds by the inspector.

"A higher boiler-pressure and a more uniform pressure than was had during test is desirable.

"The measure of economy of an engine is the amount of water used per hour per horse-power. From that point of view, your machine is far above the average.

"As the load on engine increases above 53 to 55 horse-power, the economy will decrease, unless the initial pressure is higher, or a higher speed of revolution is given to engine.

"The economy of engine by coal is not as good as it should be, and can be improved. 303 lbs. coal (anthracite) per hour per horse-power is high, but the fault is not with the engine, for the water used per hour per horse-power is only 24.9, and an evaporation of 10 lbs. actual would give an indicated horse-power for 2.5 lbs. coal ; the boiler should evaporate 10 lbs. water per lb. anthracite coal."

The following is an example of a very full report by Mr. Hill on the trial of a Corliss engine designed and built by Mr. Harris :

"The engine, 24" diameter of cylinder and 60" stroke of piston, is condensing, and fitted with the ordinary jet-condenser and reciprocating air-pump. The injection-water is obtained by a lift of 15' from the Mississippi River, upon the bank of which the mill stands ; and during the trial the condensing water entered the injection-pipe at a temperature near the freezing-point. The steam-valves were formerly closed by the usual weights ; but previous to the trial, vacuum dash-pots were added to insure a prompt closing of the valve when liberated from the hook. The engine is furnished with a pulley fly-wheel 20' diameter and 32" face ; driving back to the line-shaft with a 30" double leather belt.

"The exhaust of engine is closely connected to a condenser by a 10" pipe, and steam is conveyed from the boiler by a 7" pipe.

"Steam is furnished by a pair of tubular boilers set in battery, and each of the following dimensions: 60" diameter of shell, 12' long fifty 4" tubes. Each boiler is fitted with a vertical steam-dome, 30" diameter \times 36" high, and over these and joined to them by short legs is a horizontal steam-drum, 24" diameter and 14" long.

"The steam-pipe is joined by branch pipes to the side of the horizontal drum.

"The feed-water is taken from a drop-leg in the overflow-pipe from the condenser, and conducted to the suction of a single-acting plunger-pump driven from the engine by belt. Into the breeching or front smoke connection has been introduced a fuel economizer, consisting of 250' of 2½" iron pipe, through which the feed-water is forced to the boiler.

"The furnace is arranged to burn slabs and hard wood, although by the record it would appear to be well adapted for coal (the fuel used during the trial of engine). The lack of a suitable bridge wall, and the very large furnace-doors and grate-surface are not calculated for maximum economy with coal as a fuel, and it is eminently probable that with a different construction of furnace the efficiency of the boilers during the trial of engine would have been higher.

"The entire net power of engine is expended in driving the machinery of the mill, which consists of twelve run of 54" buhrs, and three run of 48" buhrs; two crushing rolls, each with 3-12" \times 30" cylinders; five rolls, each with 2-12" \times 30" cylinders, and one roll with 2-12" \times 18" cylinders. The bolting machinery consists of one chest with two reels; two chests with three reels; one chest with six reels, and one chest with eight reels; in all twenty-two bolting reels and forty-eight conveyors.

"The cleaning machinery consists of two 'cockle' machines; one 'scouring' machine; one 'separator,' and two brushing machines. Of the purifying machines there are seventeen;

and one shaking machine; four flour-packers; four stand of wheat elevators; four stand of flour elevators, and twenty-one middlings elevators; one small and two large exhaust fans.

"To this should be added the machinery of the grain elevator, which is driven by belt from the third story of the mill; and the line-shafting, connecting belts, pulleys, and gearing, forming the general machinery of the mill.

"In the following tables are given the principal measured and calculated data of engines and boilers. The clearance was not measured, but estimated at three per cent. of piston-displacement, this being the usual clearance in similar engines of like dimensions.

"The factor of horse-power due mean area, and velocity of piston for each mean effective pound pressure, has been calculated as follows: The area of a 24" piston is 452.39 sup. ins. and the area of the rod (3.375") is 8.9462 sup. ins.; and the mean area of piston is, therefore,

$$452.39 - \frac{8.9462}{2} = 447.917 \text{ sup. ins.,}$$

and the factor of horse-power

$$\frac{447.917 \times 596.166}{33000} = 8.20446.$$

"The valve functions have been measured on the diagrams. The volume of steam accounted for to release is obtained by taking the mean area (feet) of piston into the piston travel (feet) per hour to point of release, to which is added the hourly volume of clearance. The volume of steam retained by exhaust-closure is obtained by taking the mean area of piston, in feet, into the travel of piston, in feet per hour, from exhaust-closure to end of stroke, to which is added the hourly volume of clearance.

"The dimensions of boilers and fire-grates are furnished by your engineer, from which have been deduced the heating-surface, grate-surface, and calorimeter of tubes, and ratios of

heating to grate-surface, and grate-surface to cross-section of tubes.

Dimensions of Engine.

| | |
|-------------------------------------------------------|--------------------|
| Diameter of cylinder..... | 24 inches. |
| Stroke of piston..... | 60 " |
| Revolutions per minute during trial..... | 59.616. |
| Piston speed " " " " | 596.166 feet. |
| Factor of H. P. due area and velocity of piston | 8.204. |
| Piston stroke to release in parts of stroke.. | 99.370. |
| " " exhaust closure in parts of stroke | 6.067. |
| Clearance (estimated) in parts of stroke.... | 3.000 |
| Volume of steam to release per hour..... | 115,038.04 cu. ft. |
| " " retained by cushion per hour | 10,189.02 cu. ft. |
| Diam. of air-pump..... | 12 inches. |
| Stroke " " | 15 " |
| Diameter of driving-pulley..... | 20 feet. |
| Face " " " | 32 inches. |
| Weight " " " | 40,000 pounds. |

Dimensions of Boilers.

| | |
|-----------------------------------|--------------------|
| Number..... | 2. |
| Diameter of shells..... | 60 inches. |
| Length " " | 12 feet. |
| Tubes, each boiler..... | 50-4 inches. |
| Heating-surface shells (2). | 250.56 |
| " " tubes (100)..... | 1,245.64 |
| " " heads (4)..... | 40.72 |
| " " total..... | 1,536.92 sup. ft. |
| Grate " | 51.75 sup. ft. |
| Calorimeter of flues | 1,256.64 sup. ins. |
| Heating to grate-surface..... | 29.70 |
| Grate-surface to calorimeter..... | 5.93 |

"The trial of engine for economy of performance and trial of boilers for evaporative efficiency were made simultaneously; all preparations having been completed, the trial began at 9.15 a.m., and terminated at 7.15 p.m.; duration of trial, ten hours.

"The test of boiler efficiency was with coal.

"The load was that usually carried in the daily operation of the mill, and through the care of the chief miller, was held quite uniform during the ten-hours' run. It is possible that the mean power developed is slightly greater than usual, from the fact that the operatives were cautioned to avoid breaks in the load; and that they obeyed the injunction is best attested by the indicator-diagrams, which exhibit but slight variations in the load during the economy-trial.

"The diagrams were taken by independent indicators, one to each end of cylinder. Forty (40) pound springs were used, and the drums were moved by well-constructed bell-cranks and reciprocating connections hung on a stout gallows frame. The joints of the levers and connections were carefully made, and means were provided to take up wear and avoid lost motion.

"The strings on the indicator-barrels were only long enough to couple with the pins on the short stroke-reciprocating-bar, and the recoil-springs were adjusted as nearly as possible to the same tension. The length of diagrams was uniformly 4".78.

"During the trial a pair of diagrams were taken regularly every fifteen minutes, making eighty-two diagrams; from which have been obtained the initial pressure in cylinder, piston-stroke to cut-off, ratios of expansion by pressure and by volumes, terminal pressure, counter pressure at mid-stroke, utilization of vacuum, and mean effective pressure on the piston, from which is obtained the mean power developed.

"The vacuum in the condenser and the pressure in the boilers were taken from gauges in the engine-room regularly every fifteen minutes.

“The temperature of water to the condenser was taken in the river at the mouth of the injection-pipe. The temperature of overflow from the condenser was taken in the measuring-tank. The temperature of feed to the boiler was taken in the feed-pipe near the check-valves.

“The water delivered to the boilers was measured in the following manner: Two oil-barrels were carefully washed inside and placed on the same level in the engine-room; to the bottoms of these was connected, by branch pipes, the suction-pipe of pump, each branch being provided with an open way-cock to shut off the flow when the level had been reduced to the lowest gauge-point. The pipe from the hot-well to the pump was cut and carried out over the barrels; a connection made by branches to each barrel, and a stop-valve in each branch regulated the flow of water into the tanks. The tanks or barrels were numbered 1 and 2, and were alternately filled to the overflow notch in the rim, and emptied to the centre of the branch pipe in the side of barrel, and the contents discharged into the pipe leading to the pump.

“Whilst the No. 1 barrel was running out, the No. 2 barrel was filling with water from the hot-well; and directly the first barrel was emptied to the lower gauge-point, it was turned off and the second barrel turned on; and so on during the entire trial, the empty barrel being shut off before the full one was turned on, to prevent transfer of water from the full to the empty barrel. Directly each barrel of water was turned on, the time was entered in the log, and a tally made by the assistant in charge of the tanks. From time to time my record of tanks discharged was compared with the assistant's tally, to avoid error in the count.

“After the trial, the capacity of each tank was determined by filling to the overflow notch, noting temperature, drawing off to the lower gauge-point, and weighing. The temperatures of the tanks of water discharged into the suction-pipe of feed-pump having been regularly noted during the trial, the weight of water delivered to the boiler was deduced from the number

of tanks discharged into the weight of tanks at mean observed temperature.

"The calorimeter-tests of water entrained were made by drawing off from the steam-drum, near the pipe to the engine, a given weight of evaporation, and condensing it in a given weight of water, noting the temperature of the water before and after the steam was turned in, and the pressure of evaporation each time an observation was made. The thermal values due the ranges of temperature and the weights of steam and water, together with the thermal values of saturated steam at observed pressures, constituted the data from which has been estimated the heat-units resident in a pound of evaporation during the trial, from which has been deduced the water entrained in the steam as 12.84 per cent. of the total water pumped into boiler. Twenty calorimeter-observations were made during the ten hours' trial.

"The revolutions of the engine are nominally 60 per minute; but from the ten hours' continuous record by counter, the mean revolutions per minute were 59.616.

"The coal fired during trial of engine was Wilmington, mined in the northern part of Illinois, and, from the evaporative efficiency developed, of very fair quality.

"The ash-pit and fires were cleaned before the trial, and the ash and clinker accumulated during the ten hours' firing weighed back dry. The non-combustible by weight constituted 7.3 per cent. of the total coal fired. Previous to commencement of run, the water-level in both boilers was marked on the glass gauges, and the fires levelled and thickness noted; the same conditions of fires and water-level obtained at the end of trial.

"In the following tables are given the observed and calculated data illustrating the performance of engine and boilers. All data from the diagrams are means of eighty-two readings, and all other data are means of forty-one readings.

"The economy of engine by steam and by coal is developed upon the mean quantities charged per hour.

Data from Trial of Engine.

| | |
|----------------------------------------------|--------------|
| Duration of trial..... | 10 hours |
| Mean pressure by boiler-gauge above atm..... | 92.876 lbs. |
| “ initial pressure above atm..... | 89.376 “ |
| “ terminal “ absolute..... | 12.018 “ |
| “ counter “ “ | 2.696 “ |
| “ cut-off in parts of stroke apparent..... | 15.560 |
| “ “ “ “ actual..... | 18.019 |
| “ vacuum by gauge..... | 26.40 inches |
| “ “ “ diagrams..... | 24.05 “ |
| “ temperature of injection..... | 33.840 |
| “ “ of hot-well..... | 92.725 |
| “ effective pressure..... | 32.9792 lbs. |
| Indicated horse-power..... | 270.5796 |
| Ratio of expansion by volumes..... | 5.549 |
| “ “ “ “ pressures..... | 8.643 |

Economy of Engine.

| | |
|--------------------------------------------------|----------------|
| Total water per hour to boilers..... | 5,037.128 lbs. |
| Water (steam) per hour to calorimeter..... | 10.000 lbs. |
| “ entrained per hour in the steam..... | 655.583 lbs. |
| Net steam per hour to engine..... | 4,371.545 lbs. |
| Steam per indicated horse-power, actual,..... | 16.156 lbs. |
| “ “ “ “ by the diagrams | 13.035 “ |
| Percentage of steam accounted for..... | 80.682 |
| Coal burned per hour..... | 535. lbs. |
| Coal per indicated horse-power per hour..... | 1.9772 lbs. |
| “ “ “ “ evaporation 9 to 1 | 1.7950 “ |
| Combustible, per indicated horse-power, per hour | 1.8328 “ |

Performance of Boilers.

| | |
|------------------------------------|-------------|
| Duration of trial..... | 10 hours |
| Pressure by gauge..... | 92.876 lbs. |
| Temperature of feed to heater..... | 92.725 |
| “ “ “ “ boiler..... | 114.324 |
| Elevation of feed by heater..... | 21.599 |
| Percentage of gain by heater | 1.723 |

| | | |
|--------------------------------------------------------------|-----------|------|
| Total water pumped into boilers..... | 50,371.28 | lbs. |
| “ “ entrained in the steam (12.84 %).... | 6,467.70 | “ |
| “ steam furnished..... | 43,903.58 | “ |
| “ coal fired..... | 5,350 | “ |
| “ non-combustible weighed back..... | 390 | “ |
| “ combustible..... | 4,960 | “ |
| Steam per pound of coal..... | 8.206 | “ |
| “ “ combustible..... | 8.852 | “ |
| “ “ “ coal from temp. of 212° } and pres. of atm. } | 9.639 | “ |
| Steam per square foot of heating-surface..... | .022 | “ |
| Coal “ “ “ “ grate-surface..... | 10.300 | “ |
| Percentage of ash in coal..... | 7.3 | |
| Coal burned during trial, Wilmington (Illinois). | | |

“ During the economy-trial of engine, the flour manufactured was, by the miller's report, 217 barrels high grade, and 2 per cent. added for low grade, or 221.34 barrels produced in ten hours. The mean indicated power of engine was 270.56 horsepower, and the hourly expenditure of power per barrel of flour produced was $\frac{270.56}{221.34} = 12.224$ H. P.

“ The coal burned for whole trial was 5350 pounds, and coal per barrel of flour produced becomes $\frac{5350}{221.34} = 24.198$ pounds.

“ Whilst the experiments (firing, slabs and hard wood) were in progress, the engine was indicated for distribution of the power in the mill.

“ The first (*A*) load was with all the machinery on, and operating under the ordinary conditions. The second (*B*) load was with all the machinery on except the machinery in elevator-building. The third (*C*) load was with all the machinery on except the flour-packers. The fourth (*D*) load was with all the machinery on except the cleaning machinery and flour-packers. The fifth (*E*) load was with all the machinery on except the crushing-rolls. The sixth (*F*) load was with all the machinery on except the purifiers. And the seventh (*G*) load was with all the machinery on except the grinding-buhrs.

" The changes of load were made quickly, in order to preserve the conditions of ordinary performance in the special machinery driven; and the power developed for each load has been estimated from six diagrams, three from each end of cylinder.

" The indicated loads were as follows:

| | | |
|--------------------------------|---------|-------|
| First load (<i>A</i>)..... | 267.503 | H. P. |
| Second load (<i>B</i>)..... | 262.585 | " |
| Third load (<i>C</i>)..... | 363.706 | " |
| Fourth load (<i>D</i>)..... | 250.726 | " |
| Fifth load (<i>E</i>)..... | 246.740 | " |
| Sixth load (<i>F</i>)..... | 242.645 | " |
| Seventh load (<i>G</i>)..... | 117.149 | " |

" Each of these loads is made up of the friction of engine in all parts; extra friction of engine due to the load, friction of all the driving machinery in the mill, and power required to drive the special machinery, including friction; in like manner the differences between the maximum load and reduced loads nearly represent the power required to drive the special machinery not on, including its own friction.

" Hence the difference between the maximum load and lesser loads represents slightly more than the power actually absorbed by the special machinery not driven.

" The distribution of the power in the mill is thus as follows:

| | |
|-------------------------------------------------------------------|---------|
| Total indicated power of engine load (<i>A</i>) | 267.503 |
| Friction of engine alone..... | 16.409 |
| Extra friction due load..... | 12.544 |
| Grinding-buhrs..... | 150.354 |
| Cleaning machinery..... | 12.980 |
| Elevator..... | 4.918 |
| Crushing-rolls..... | 20.763 |
| Bolting-reels, conveyors, fans, and } general machinery..... } | 21.860 |
| Middlings-purifiers..... | 23.868 |
| Flour-packers..... | 3.797 |
| | 267.503 |

"I have attached to this report one pair of diagrams taken during the trial, and numbered 14, upon which have been drawn theoretic curves from the terminal pressures and points of release. The lack of coincidence between the actual and theoretic curves is due, in my opinion, to a slight leak past the piston or out of the exhaust; in all probability, the latter. The engine being very new, and a certain amount of wear being requisite to make good joints between the valve faces and seats, it is probable that this leak will in time remedy itself. However, it does not appear that the failure of the diagram to satisfy the conditions of theory has any marked effect on the economy of the engine, for the actual consumption of steam and coal per indicated horse-power per hour are the least I have ever obtained from a single-cylinder engine."

The preceding illustrations exhibit the methods of report and the results of trial of good examples of ordinary practice with the common simple engine of moderate size and under usual working conditions.

As illustrating what can be done with a good compound stationary engine, the following data are presented, as given the Author by Mr. Corliss a short time before his death, as the results obtained from a compound condensing-engine of about 500 I. H. P., driving a cotton-mill:

RESULT OF A SEVEN DAYS' TRIAL OF THE CORLISS COMPOUND ENGINE
AT NOURSE MILL.

Commencing Oct. 15th, and ending Oct. 22d, 1884.

| | STARTING FIRES. | | | Coal while run- ning. | Wood used during run- ning time. | Value of same at 40 %. | Total new fuel fed to fur- nace each day. | Cinders used during day. |
|----------------------------------------|-----------------|------------------------------|-------|--------------------------------|-------------------------------------------------|------------------------------|-------------------------------------------------------------|-----------------------------------|
| | Wood. | Value at 40 % of Coal. | Coal. | | | | | |
| Wed., Oct. 15, 1884.. | 1000 | 400 | 2647 | 6838 | 200 | 80 | 9965 | 1671 |
| Thur, " 16, " .. | 833 | 333 | 2535 | 7200 | | | 10068 | |
| Fri., " 17, " .. | 30 | 12 | 2800 | 7200 | | | 10012 | |
| Sat., " 18, " .. | | | 3200 | 3600 | | | 6800 | |
| Mon., " 20, " .. | 800 | 320 | 3200 | 7200 | | | 10720 | |
| Tue., " 21, " .. | 10 | 4 | 3200 | 6904 | | | 10108 | 3018 |
| Wed., " 22, " .. | | | 3200 | 4300 | 30 | 12 | 7512 | |
| Totals and averages for 7 days..... | 2673 | 1069 | 20782 | 43242 | 230 | 92 | 65185 | |

| | Total fuel fed to fur- nace each day. | Cinders found at end of day. | Total Fuel used. | Run- ning Time. | Fuel used per hour. | Indi- cated Horse- power. | Fuel per hour per I. H. P. | Aver- age Rev. |
|----------------------------------------|------------------------------------------------------|---------------------------------------|------------------------|-----------------------|------------------------------|------------------------------------|----------------------------------------|----------------------|
| Wed. Oct. 15 1884... | 9965 | 863 | 9102 | 11 H. 44 M. | 775.76 | 498.93 | 1.55 | 56.96 |
| Thur. " 16 " ... | 10068 | 685 | 9383 | 11 " 41 " | 803.13 | 496.37 | 1.61 | 57.10 |
| Fri. " 17 " ... | 10012 | 730 | 9282 | 11 " 40 " | 795.64 | 495.09 | 1.60 | 57.08 |
| Sat. " 18 " ... | 8471 | 729 | 7742 | 9 " 44 " | 795.41 | 483.80 | 1.64 | 57.46 |
| Mon. " 20 " ... | 10720 | 742 | 9978 | 11 " 42 " | 852.82 | 509.46 | 1.67 | 56.84 |
| Tue. " 21 " ... | 10108 | 717 | 9391 | 11 " 40 " | 804.98 | 504.24 | 1.59 | 56.99 |
| Wed. " 22 " ... | 10530 | 455 | 10075 | 11 " 41 " | 862.36 | 501.94 | 1.71 | 57.17 |
| Totals and averages for 7 Days..... | | | | 79 H. 52 M. | 812.38 | 499.13 | 1.62 | 57.10 |

General Summary of 7 Days' Trial of Nourse Mill Engine.

Commencing Oct. 15th, and ending Oct. 22d, 1884.

Fires were started on clean grates Wed., Oct. 15th, 1884.

Wood 2903 lbs. Equivalent value in Coal at 40

per cent., 1161 lbs.

Coal for starting fires, 20782 "

Coal used while running, 43242 "

Total amount of fuel fed to furnaces, 65185 "

Deduct cinders found at end of run, 455 lbs.; esti-
mated at 80 per cent. value of new coal, . . . 364 "

Total fuel used during the seven days, 64821 "

Running time, 79 H. 52 M.

Fuel per hour, 811.62 lbs.

Average indicated horse-power from 636 cards, . . 499.13

Fuel per hour per indicated H. P., 1.626 lbs.

Average revolutions per minute, 57.1

Per cent. of ash and clinker, 11.9

The results of a series of trials of a single-acting compound engine, as given in the following table, illustrate well the system and the conciseness which characterize the work of the expert engineer, as well as the efficiency attainable with this class of engines when well designed, well constructed, and operated under favorable conditions:

STEAM PER INDICATED H. P. SINGLE-ACTING COMPOUND ENGINE.

| NON-CONDENSING. | | | | | CONDENSING. | | | |
|------------------|---------|----------|----------|--------------|------------------|----------|---------|---------|
| Boiler-pressure. | | | | Horse-power. | Boiler-pressure. | | | |
| 60 lbs. | 80 lbs. | 100 lbs. | 120 lbs. | | 120 lbs. | 100 lbs. | 80 lbs. | 60 lbs. |
| | | | 22.6 | 210 | 18.4 | | | |
| | | 23.0 | 21.9 | 170 | 18.1 | 18.8 | | |
| | 24.9 | 23.6 | 22.2 | 140 | 18.2 | 18.5 | 20.0 | |
| | 25.7 | 23.9 | 22.2 | 115 | 18.2 | 18.6 | 19.6 | 20.5 |
| 26.9 | 25.2 | 24.9 | 22.4 | 100 | 18.3 | 18.6 | 19.7 | 20.3 |
| 27.7 | 25.2 | 25.1 | 24.6 | 80 | 18.3 | 18.6 | 19.9 | 20.1 |
| 30.3 | 28.7 | 29.4 | 28.8 | 50 | 20.4 | 20.8 | 20.7 | 20.4 |

The engine was of 14 and 24 inches diameter of cylinder, 14 inches stroke of piston, and unjacketed. All steam passing into and through the engine was weighed and measured. Gauge-pressures are given. This engine is usually rated at 150 H. P. The steam was probably dry, but not superheated.

95. Tabulated Deductions and General Conclusions from engine-trials, as above illustrated, should usually be presented in as concise and legible form as possible, and so arranged as to make it easy to interpret the data and to verify the results both as to facts and reasoning. Thus, one of the most complete investigations ever made in this field was that of Mr. Willans on the efficiency and wastes of his triple-expansion engine.* He describes in his report a series of economy-trials, non-condensing, made with one of his central-valve engines, with one crank, having three cylinders in line. By removing one or both of the upper pistons, the engine could be easily changed into a compound or into a simple engine at pleasure.

Check-trials were made by Mr. MacFarlane Gray, Prof. Kennedy, Mr. Druitt Halpin, Professor Unwin, and Mr. Wilson Hartnell. The work theoretically due from a given quantity of steam at given pressure, exhausting into the atmosphere, was first considered.

By a formula of Mr. MacFarlane Gray, which agreed with the less simple formulas of Rankine and Clausius, the weights of steam required theoretically per indicated horse-power were ascertained.

A description is given of the main series of trials, of the ap-

* Proc. Brit. Inst. C. E., Mch. 13, 1888. Sci. Am. Supp., May 20, 1888.

pliances used, and of the means taken to insure accuracy. The missing quantity of feed-water at cut-off, which, in the simple engine-trials, rose from 11.7 per cent. at 40 lbs. absolute pressure to nearly 30 per cent. at 110 lbs., and at 90 lbs. was 24.8 per cent., was at 90 lbs. only 5 per cent. in the compound trials. In the latter, at 160 lbs., it increased to 17 per cent., but on repeating the trial with triple expansion, it fell to 5.46 per cent. or to 4.43 per cent. in another trial not included in the table.

The compound engine must always give a smaller diagram, considered with reference to the steam present at cut-off, than a simple engine, and a triple a smaller diagram than a compound engine. But even at 80 lbs. absolute, the compound engine had an advantage, not only from reduced initial condensation, but from less loss from clearance, and from reduced leakage. These gains became more apparent with increasing wear. The greater surface in a compound engine had not the injurious effect sometimes attributed to it, and the author showed how much less the theoretical diagram was reduced by the two small areas taken out of it in a compound engine than by the single large area abstracted in a simple engine. The trials completely confirmed the view that the compound engine owed its superiority to reduced range of temperature. At the unavoidably low pressures of the trial, the losses due to the added passages, etc., almost neutralized the saving in initial condensation; but with increased pressure—say to 290 lbs. absolute—there would be considerable economy. The figures of these trials showed that the loss of pressure due to passages was far greater with high- than with low-pressure steam, and that pipes and passages should be proportioned with reference to the weight of steam passing, and not for a particular velocity merely.

After comparing the data of initial condensation in cases where the density of steam, the area of exposed surface, and the range of temperature were all variables, with other cases (1) where the density was constant and (2) where the surface was constant, the author concluded that, at four hundred revolutions per minute, the amount of initial condensation depended chiefly on the range of temperature in the cylinder, and not

upon the density of the steam or upon the extent of surface, and that its cause was probably the alternate heating and cooling of a small body of water retained in the cylinder. The effect of water intentionally introduced into the air-cushion cylinder showed how small a quantity of water retained in the cylinder would account for the effects observed. At lower speeds, surface might have more influence. The favorable economical effect of high rotative speed was very apparent.

Starting with 40 indicated horse-power, 130 lbs. absolute pressure, four expansions, and a consumption of 20.75 lbs. of water, the plan of varying the expansion, as compared with throttling, showed a gain of about 7 per cent. at 30 indicated horse-power, but of a very small percentage when below half-power. If the engine had an ordinary slide-valve, the greater friction, added to irregular motion, would probably neutralize the saving; while if the engine were one in which initial condensation assumed more usual proportions, the gain would be probably on the side of variable pressure. The diagrams showed that the missing quantity became enormously large as the expansion increased. Judging only by the feed-water accounted for by the indicator, the automatic engine appeared greatly the more economical, but actual measurement of the feed-water disproved this. The position of the automatic engine was relatively more favorable when simple than when compound.

The tabulated figures are given on page 349.

M. Delafond, testing a single-cylinder Corliss engine, built at Creusot, with similar care and thoroughness, comes to the following conclusions as fairly deducible from the data so obtained:*

(1) The effective work is equal to

$$T_e = -\alpha + \beta T_i;$$

where T_i is the indicated work; but the coefficients α and β are not absolutely constant, varying with the pressure of steam.

* *Essais Effectuées sur une Machine Corliss, etc.*; Paris, Dunod, Editeur, 1884; p. 60.

TRIALS OF TRIPLE EXPANSION ENGINES.

| Intended mean admission pressure....Lb. | 40 | 90 | | 110 | | 130 | 150 | | 160 | | 170 |
|--------------------------------------------------------------------------------------------------------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|
| Simple. Compound or Triple. | S. | S. | C. | S. | C. | C. | C. | T. | C. | T. | T. |
| Actual mean admission pressure....Lb. | 40.87 | 92.65 | 87.54 | 106.3 | 109.3 | 130.6 | 149.9 | 151.9 | 158.5 | 158.1 | 172.5 |
| Percentage ratio of actual mean pressure, referred to low-pressure piston, to theoretical mean pressure..... | 98.26 | 100 | 91.3 | 100.7 | 94.8 | 94.2 | 94.6 | 84.54 | 95.9 | 85.3 | 85.2 |
| Ind. horse-power..... | 16.51 | 31.61 | 29.14 | 33.5 | 33 | 36.31 | 38.59 | 35.69 | 39.55 | 35.36 | 38.45 |
| Feed-water actually used per indicated H. P. H.— | | | | | | | | | | | |
| Simple... ..Lb. | 42.76 | 26.89 | | 26 | | | | | | | |
| Compound...Lb. | | | 24.16 | .. | 21.37 | 20.35 | 19.45 | | 19.19 | | |
| Triple... ..Lb. | | | | | | | | | | | |
| Steam required theoretically per I. H. P. H.....Lb. | 34.67 | 19.24 | 19.86 | 17.9 | 17.65 | 16.25 | 15.23 | 19.68 | | 19.19 | 18.45 |
| Percentage efficiency. | 81.1 | 71.5 | 82.2 | 68.8 | 82.5 | 80.0 | 78.3 | 77 | 77.4 | 77.6 | 77.8 |
| Percentage of feed-water missing at cut-off in high-pressure cylinder..... | | | | | | | | 5.33 | ... | 6.84 | 5.01 |
| Ditto intermediate cylinder..... | | | 5 | | 9.5 | 11.7 | 15.1 | 14.84 | 17 | 12.06 | 15.33 |
| Ditto low-pressure cylinder..... | 11.7 | 24.8 | 15.2 | 29.56 | 16.25 | 19.1 | 20.6 | 22.12 | 21.3 | 22.11 | 24.21 |
| Percentage of feed-water missing at end of stroke in low-pressure cylinder.... | 10.4 | 18.83 | 14.25 | 21.53 | 16.59 | 17.55 | 20.69 | 18.01 | 19.55 | 18.81 | 19.25 |

(2) The best economy was measured under the following conditions :

With condenser;

“ jacket;

“ moderate pressure;

“ “ expansion.

In the best cases, the expenditure of steam, per effective and per indicated horse-power, per hour, was respectively 9^k.50 and 7^k.75 (about 17 and 21 pounds).

(3) The steam-jacket is advantageous for high ratios of expansion and high pressures; its effect decreases with reduced pressure and expansions, and becomes insignificant at low pressures and small ratios of expansion.

(4) Compression is useful in non-condensing engines, and

the more so as the final pressure is made to approximate initial steam-pressure.

(5) The best results were obtained in these cases at 120 to 170 I. H. P. At higher powers the cost in steam rapidly rises. Above 175 horse-power, the condenser is of no advantage, and high compression and the use of a good feed-water heater are advisable.*

(6) Initial condensation increases with increase of pressure, and lessens with diminishing expansion, becoming insignificant at full-stroke. The jacket reduces this loss; but the presence or absence of the condenser has no effect, this cylinder condensation being a result of expansion.

(7) The cylinder condensations and re-evaporations are of complex character; the jacket increases the re-evaporation; pressure of $3\frac{1}{2}$ or $4\frac{1}{2}$ atmospheres gave largest evaporations.

(8) Moderate pressures are best; † as they give small initial condensation and considerable re-evaporation. ‡

(9) High piston-speed and the use of steam in the jacket of higher pressure than in the engine are advisable.

(10) Working non-condensing at pressures of $3\frac{1}{2}$ to $5\frac{1}{2}$ atmospheres, with small expansion; the permanent presence of water in the cylinder could not be detected.

The following tables, prepared by M. Isherwood from the data obtained by a committee of the Société Industrielle de Mulhouse at a series of trials of a condensing compound engine, illustrate well the fulness sometimes considered desirable in such cases.§ But it will be observed that even here such important data as the ratios of expansion, release, and compression, and the quantity of cylinder condensation were not obtained.

* This conclusion should, in the view of the Author, be based on a limit of steam-pressure (perhaps above 75 lbs. by gauge).

† For single cylinder or simple engines.

‡ The Author questions the logic of this deduction.

§ Bulletin de la Soc. Ind. de Mulhouse, Jan.-Feb. 1880. Journal Franklin Institute, Oct. 1885.

EXPERIMENTS ON A CONDENSING COMPOUND ENGINE, INDUSTRIAL SOCIETY
OF MULHOUSE.

| | | July 8, 1879. Morning. | July 8, 1879. Afternoon. |
|--------------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------|---------------------------|-----------------------------|
| Total Quantities. | Duration of the experiments, in consecutive hours and decimals of an hour..... | 3.00222 | 3.24139 |
| | Total number of double strokes made by the pistons of the engine..... | 16211. | 17253. |
| | Total number of pounds of feed-water pumped into the boiler..... | 3248.068561 | 4265.391060 |
| | Total number of pounds of condensing-water admitted to the condenser..... | 74038.567197 | 78893.841895 |
| | Total number of pounds of condensing-water and water of steam condensation withdrawn from the condenser..... | 77253.139183 | 84499.777450 |
| Engine. | Steam, pressure in boiler, in pounds per square inch above the atmosphere..... | 91.694 | 92.164 |
| | Pressure in the condenser, in pounds per square inch above zero..... | 1.810 | 1.907 |
| | Number of double strokes made per minute by the pistons of the engine..... | 89.994448 | 88.711972 |
| | Position of the throttle-valve..... | Wide open. | Wide open. |
| | Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off..... | 0.25 | 0.42 |
| | Fraction completed of the stroke of the piston of the small cylinder when the steam was released..... | 0.98 | 0.98 |
| | Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned..... | 0.925 | 0.925 |
| | Fraction completed of the stroke of the piston of the large cylinder when the steam was cut off..... | 0.45 | 0.45 |
| | Fraction completed of the stroke of the piston of the large cylinder when the steam was released..... | 0.91 | 0.91 |
| | Fraction completed of the stroke of the piston of the large cylinder when the steam was cushioned..... | 0.75 | 0.75 |
| | Number of times the steam was expanded..... | 9.6444 | 6.2569 |
| | Atmospheric pressure, in pounds per square inch above zero..... | 14.222813 | 14.222813 |
| | | | |
| Water. | Number of pounds of feed-water pumped into the boiler per hour..... | 1081.838922 | 1315.914179 |
| | Number of pounds of condensing-water admitted to the condenser per hour..... | 24661.273057 | 24648.018873 |
| | Temperature in degrees Fahrenheit of the condensing-water when admitted to the condenser..... | 48.001 | 48.010 |
| | Number of pounds of condensing-water and water of steam condensation withdrawn from the condenser per hour..... | 25732.004711 | 26068.994305 |
| | Temperature, in degrees Fahrenheit, of the condensing-water and water of steam-condensation when taken from condenser..... | 86.932 | 96.728 |
| Steam-pressures in Small Cylinder per Indicator. | Number of Fahrenheit units of heat consumed per hour..... | 1220557.870799 | 1471790.924665 |
| | Pressure on piston of small cylinder at commencement of its stroke, in pounds per square inch above zero..... | 99.787254 | 102.966053 |
| | Pressure on piston of small cylinder at point of cutting off the steam, in pounds per square inch above zero.... | 88.280999 | 91.459798 |
| | Pressure on piston of small cylinder at the end of its stroke, in pounds per square inch above zero..... | 33.423600 | 45.184067 |
| | Mean back-pressure against piston of small cylinder during its stroke, in pounds per square inch above zero.... | 35.343690 | 46.449021 |
| | Back-pressure against piston of small cylinder at the point where the cushioning began, in pounds per square inch above zero..... | 34.846000 | 42.614000 |
| | Indicated pressure on the piston of the small cylinder, in pounds per square inch..... | 27.855379 | 30.678629 |
| | Net pressure on the piston of the small cylinder, in pounds per square inch..... | 24.227924 | 27.046174 |
| | Total pressure on the piston of the small cylinder in pounds per square inch..... | 60.200000 | 73.300000 |
| | | | |

| | | July 8, 1879. Morning. | July 8, 1879. Afternoon. |
|-----------------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------------|-----------------------------|
| Steam-pressures in Large Cylinder per Indicator. | Pressure on piston of large cylinder at commencement of its stroke, in pounds per square inch above zero..... | 34.490000 | 43.200000 |
| | Pressure on piston of large cylinder at point of cutting off the steam, in pounds per square inch above zero.... | 21.156000 | 27.740000 |
| | Pressure on piston of large cylinder at the end of its stroke, in pounds per square inch above zero | 9.600000 | 12.561000 |
| | Mean back-pressure against piston of large cylinder during its stroke, in pounds per square inch above zero.. | 3.756000 | 3.756000 |
| | Back-pressure against piston of large cylinder at the point where the cushioning began, in pounds per square inch above zero..... | 2.355000 | 2.355000 |
| | Indicated pressure on the piston of the large cylinder, in pounds per square inch..... | 17.006220 | 22.300000 |
| | Net pressure on the piston of the large cylinder, in pounds per square inch.... | 15.747273 | 21.041053 |
| | Total pressure, in pounds per square inch above zero, on the annular surface of the piston of the large cylinder remaining after subtracting from the area of that piston the area of the piston of the small cylinder..... | 19.556220 | 24.850000 |
| | Indicated horses-power developed in the small cylinder.. | 23.135964 | 25.113672 |
| | Net horses-power developed in the small cylinder.... | 20.123093 | 22.143736 |
| | Total horses-power developed in the small cylinder..... | 50.000578 | 60.013510 |
| Horses-power. | Total horses-power developed in the small cylinder by the expanded steam alone..... | 28.101654 | 28.560045 |
| | Indicated horses-power developed in the large cylinder.. | 40.738193 | 52.658120 |
| | Net horses-power developed in the large cylinder..... | 37.722401 | 49.685305 |
| | Total horses-power developed in the large cylinder by the annular surface of its piston remaining after subtracting from the area of that piston the area of the piston of the small cylinder | 30.603792 | 38.333020 |
| | Total horses-power developed in the large cylinder by the expanded steam alone..... | 32.491077 | 40.194311 |
| | Aggregate indicated horses-power developed by the engine | 63.874157 | 77.771792 |
| | Aggregate net horses power developed by the engine.... | 57.845494 | 71.829041 |
| | Aggregate total horses-power developed by the engine, exclusive of cushioning..... | 80.604370 | 98.347430 |
| | Aggregate total horses-power developed by the expanded steam alone in both cylinders inclusive of cushioning..... | 60.592731 | 68.754356 |
| | Horses-power developed by the engine at the friction brake applied to the wheel on the engine shaft | 55.717593 | 67.499569 |
| | Per centum of the total horses-power developed by the engine utilized as indicated horses-power..... | 79.24423 | 79.07862 |
| | Per centum of the total horses-power developed by the engine utilized as net horses-power..... | 71.76471 | 73.03601 |
| | Per centum of the total horses-power developed by the engine utilized as horses-power at the friction brake... | 69.12477 | 68.63379 |
| Economic Results. | Number of pounds of feed-water consumed per hour per indicated horse-power | 16.937819 | 16.920199 |
| | Number of pounds of feed-water consumed per hour per net horse-power | 18.703080 | 18.320085 |
| | Number of pounds of feed-water consumed per hour per total horse-power developed by the engine..... | 13.422212 | 13.380260 |
| | Number of pounds of feed-water consumed per hour per horse-power developed by the engine at the brake.... | 19.417366 | 19.495149 |
| | Number of Fahrenheit units of heat consumed per hour per indicated horse-power | 19108.790286 | 18924.482602 |
| | Number of Fahrenheit units of heat consumed per hour per net horse-power | 21100.310264 | 20490.193161 |
| | Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the engine | 14142.576895 | 14965.219982 |
| | Number of Fahrenheit units of heat consumed per hour per horse-power developed by the engine at the brake... | 21906.148580 | 21804.449220 |

| | | July 8, 1879. Morning. | July 8, 1879. Afternoon. |
|------------------------------------------------------------------------------------------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------------|-----------------------------|
| Weight of Steam accounted for by the Indicator. | Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there..... | 607.871477 | 1005.726509 |
| | Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there..... | 879.050774 | 1170.906554 |
| | Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanded steam alone in that cylinder..... | 76.530424 | 78.900853 |
| | Sum of the two immediately preceding quantities..... | 955.581198 | 1249.807407 |
| | Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there..... | 805.995933 | 1047.498116 |
| | Number of pounds of steam condensed per hour in the small and large cylinders, and in the receiver to furnish the heat transmuted into the total horses-power developed in those cylinders by the expanded steam alone after the closing of the cut-off valve on the small cylinder..... | 162.065479 | 185.609642 |
| | Sum of the two immediately preceding quantities..... | 968.061412 | 1233.197758 |
| Difference between the Weight of Water Vaporized in the Boiler and the Weight of Steam accounted for by the Indicator. | Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam..... | 474.017445 | 310.187670 |
| | Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the point of cutting off the steam..... | 43.813874 | 23.572029 |
| | Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston..... | 126.307724 | 66.106772 |
| | Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the small cylinder at the end of the stroke of its piston..... | 11.674740 | 5.023638 |
| | Difference in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston..... | 113.827510 | 82.716421 |
| | Difference in per centum of the weight of water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the large cylinder at the end of the stroke of its piston..... | 10.521183 | 6.285852 |
| Cylinder-pressures Reduced to Large Cylinder alone. | Indicated pressure, in pounds per square inch, that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio of the area of the small to that of the large cylinder, and the quotient added to the experimental indicated pressure on the piston of the large cylinder..... | 26.664363 | 32.935300 |
| | Net pressure in pounds per square inch, that would be on the piston of the large cylinder, similarly calculated to the immediately above..... | 24.147684 | 30.418625 |
| | Total pressure, in pounds per square inch above zero, that would be on the piston of the large cylinder, corresponding to the total horses-power developed by the engine..... | 33.648414 | 41.648801 |
| | Back-pressure, in pounds per square inch above zero, that would be against the piston of the large cylinder similarly calculated to the above..... | 6.984051 | 8.713501 |

This engine was two-cylinder compound, the cylinders placed side by side, an independent cut-off slide on the small cylinder, adjusted by a loaded governor, but with no cut-off valve on the low-pressure cylinder. The small cylinder was jacketed on its sides and the large cylinder half-jacketed. The dimensions were 112 and 18.9 inches (30 and 50 cm.), and 18.9 inches (50 cm.) stroke of piston.

In the table the "indicated" power is that shown by the diagrams: the "net" power is the indicated power less that expended in overcoming the friction of the engine; the "total" power is that shown by the indicator, including that expended in overcoming back-pressure; *i.e.*, it is given by the diagram measured down to the perfect vacuum line, and is the total mechanical effect produced by the fuel.

A trial of a small compound engine made under the direction of the Council of the Society of Arts, to determine its value as a motor for electric-lighting purposes, gave the following data.* The engine was of 5".24 and 8".98 diameter of cylinder, and of 14" stroke of piston.

The data obtained from the log of the second trial being plotted, gave a graphical representation of the whole course of the trial, thus:

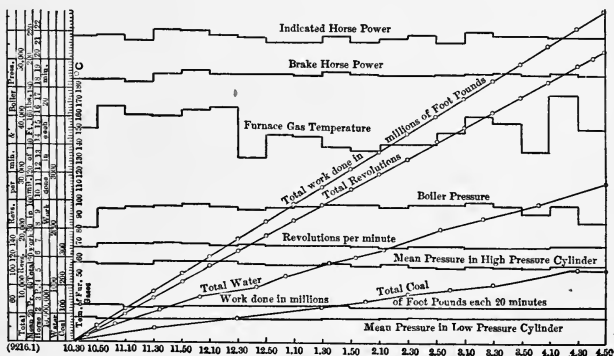


FIG. 121.—GRAPHICAL LOG.

* Journal Society of Arts, Feb. 15, 1889, p. 235.

TEST OF A SMALL COMPOUND ENGINE.

| | Date..... | Sept. 26 A ₁ | Sept. 28 A ₂ | Oct. 1 B | Oct. 1 C |
|----|-------------------------------------------------------------------------|----------------------------|----------------------------|-------------|-------------|
| 1 | Trial | | | | |
| 2 | Duration..... | 6.43 hours | 6.27 hours | 3.12 hours | 1 hour |
| 3 | Power..... | Full | Full | Half | Empty |
| 4 | Revolutions per minute..... | 140.48 | 137.35 | 138.10 | 144.20 |
| 5 | Mean initial pressure, high-pressure cylinder..... | 176.6 | 176.8 | 113.0 | |
| 6 | Mean ratio of expansion | | | | |
| 7 | Mean effective pressure, high-pressure cylinder..... | 52.93 | 54.98 | 33.25 | 11.32 |
| 8 | Mean effective pressure, low-pressure cylinder..... | 17.56 | 16.78 | 8.92 | -1.19 |
| 9 | Indicated H. P. effective pressure, high-pressure cylinder | 11.14 | 11.30 | 6.85 | +2.40 |
| 10 | Indicated H. P. effective pressure, low-pressure cylinder..... | 10.98 | 10.26 | 5.47 | -0.12 |
| 11 | Indicated H. P. Total..... | 22.12 | 21.56 | 12.32 | 2.28 |
| 12 | Brake load net..... | 288.8 | 288.0 | 147.6 | |
| 13 | Brake H. P. | 19.44 | 18.95 | 9.76 | |
| 14 | Mechanical efficiency..... | 0.879 | 0.879 | 0.792 | |
| 15 | Indicated H. P. in driving engine..... | 2.68 | 2.61 | 2.56 | 2.28 |
| 16 | BOILER. | | | | |
| 17 | Mean boiler-pressure (above atmosphere)..... | 191.35 | 187.98 | 120.10 | |
| 18 | Atmospheric pressure for the day | 14.9 | 14.8 | 14.8 | |
| 19 | Boiler-pressure (absolute)..... | 206.25 | 202.78 | 134.9 | |
| 20 | Temperature of boiler-steam..... | 384.3° | 382.9° | 350.1° | |
| 21 | Pounds of feed used per hour..... | [448.7] | 447.1 | 392.2 | |
| 22 | “ “ “ indicated H. P. per hour..... | [20.28] | 20.74 | 26.72 | |
| 23 | “ “ “ brake H. P. per hour..... | [23.08] | 23.59 | 33.73 | |
| 24 | Mean temperature of feed-water in tank, before | 63.0 | 63.0 | 69.9 | |
| 25 | entering coil..... | [201] | 201 | | |
| 26 | Mean temperature of feed-water before entering exhaust feed-heater..... | 115.7 | 63.0 | | |
| 27 | Temperature of chimney gases (Fahr.)... | 355.4° | 304.4° | 369.1° | |
| 28 | Coal per hour..... | 39.66 | 40.70 | 27.25 | |
| 29 | “ “ “ per indicated H. P. | 1.79 | 1.89 | 2.21 | |
| 30 | “ “ “ per brake H. P. | 2.01 | 2.15 | 2.79 | |
| 31 | Pounds of water per pound of coal..... | | 10.99 | 12.08 | |
| 32 | and at 212° F..... | | 11.71 | 12.76 | |

DISTRIBUTION OF HEAT A₂.

| | Thermal Units. | Percent-ages. |
|----------------------------------------------------------------------------------------------------------|----------------|---------------|
| Calorific value of 40.7 pounds of coal..... | 577,900 | 100 |
| Heat expended in heating and evaporating water, including heat given up by gases to coil in chimney..... | 460,300 | 79.65 |
| Heat expended in raising temperature of furnace-gases..... | 40,700 | 7.05 |
| Heat lost by radiation..... | 51,070 | 8.85 |
| Heat lost by imperfect combustion..... | 15,450 | 2.68 |
| Heat expended in evaporating moisture in coal..... | 570 | 0.10 |
| Heat lost in ash and otherwise unaccounted for..... | 9,810 | 1.67 |
| | 577,900 | 100.00 |

Engine.—The work done, 21.55 indicated h. p., corresponds to 55,300 thermal units per hour, or 12.0 per cent. of the whole heat taken up by the water. The efficiency of a perfect heat-engine working between 383° and 212° F. would be 0.205. Such an engine receiving the same amount of heat as the Paxman engine, namely, 460,300 thermal units per hour, would turn into work 93,200 thermal units per hour. The actual efficiency of the engine therefore, compared with such a perfect engine, is 59 per cent. The heat received by the engine per indicated h. p. per hour was 21,350 thermal units. The brake h. p. of the engine was 18.95 its mechanical efficiency was therefore 87.9 per cent., the indicated h. p. expended in driving the engine itself being 2.61.

Boiler and Engine.—The combined efficiency of the furnace, boiler, and engine, as represented by the consumption of fuel per horse-power, works out to 9.6 per cent., 55,300 thermal units being turned into work per hour, with an expenditure of fuel having a value of 577,900 thermal units. The coal used per indicated h. p. per hour was 1.89 pounds, and per brake h. p. per hour 2.15 pounds.

Steam per Indicator-cards.—The amount of steam shown by the indicator-diagrams (Figs. 18 and 19) was as below :

| | Percentage of whole weighed feed-water. |
|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----------------------------------------|
| Steam in h. p. cylinder at a pressure of 150 lbs. per square inch above the atmosphere, which corresponds to a point at 0.39 of the stroke, a little after cut-off in all cases..... | 65.0 |
| Steam in l. p. cylinder at a pressure of 10 lbs. per square inch above the atmosphere, which corresponds to a point at 0.67 of the stroke, well before release in all cases..... | 78.8 |

96. Examples of Trials of Portable Engines are, perhaps, best found in the annual reports of the competitive tests of such engines at the various agricultural exhibitions, where they are most frequently seen and subjected to trial. These trials are of peculiar interest, as exhibiting the fact that even the smallest engines may, by skilful design, construction, and management, be made to give admirable economy. The following are the data and the results reported at Newcastle, G. B., by the committee of judges, aided by Messrs. Sir Frederick Bramwell and W. Anderson :*

* Journal Agricult. Soc. of England, vol. xxiii, 1887.

DIMENSIONS OF ENGINES.

| | SIMPLE ENGINES. | | | | | | COMPOUND ENGINES. | | | | | |
|--------------------------------------------------------------------------|------------------------------------|----------------------|---------------------|----------------------|----------------------|-----------------------|----------------------|----------------------|----------------------|----------------------|----------------------|--|
| | 3111 | 3114 | 147 | 3108 | 3125 | 3117 | 3115 | 3113 | 3124 | 3107 | 3116 | |
| Catalogue number | | | | | | | | | | | | |
| Diameter and stroke of high-pressure cylinder in inches | 8 $\frac{3}{4}$ X 12 | 7 $\frac{1}{2}$ X 10 | 5 $\frac{1}{2}$ X 9 | 8 $\frac{1}{2}$ X 15 | 9 $\frac{1}{2}$ X 12 | 10 $\frac{1}{2}$ X 14 | 6 X 11 | 4 $\frac{1}{2}$ X 10 | 5 $\frac{1}{2}$ X 14 | 5 $\frac{1}{2}$ X 15 | 7 $\frac{1}{2}$ X 14 | |
| Diameter and stroke of low-pressure cylinder in inches | | | | | | | 9 X 11 | 9 $\frac{1}{2}$ X 10 | 9 $\frac{1}{2}$ X 14 | 9 X 15 | 12 X 14 | |
| Mean clearances and ports, high-pressure cylinder, in cubic inches | 100.7 | | 23.0 | 99.5 | 90 | | 75.0 | | 54.0 | 62.2 | | |
| Mean clearances and ports, low-pressure cylinder in cubic inches | | | | | | | 108.3 | | 128.5 | 103.0 | | |
| Intermediate space, cubic inches | | | | | | | 875 | | 517 | 427.4 | | |
| Diameter and width of fly-wheel | 48 $\frac{1}{2}$ X 6 $\frac{1}{2}$ | 53 X 6 | 40 X 5 | 60 X 7 $\frac{1}{2}$ | 60 X 7 | 60 X 7 $\frac{1}{2}$ | 48 X 6 $\frac{1}{2}$ | 53.6 | 60 X 6 | 60 X 7 $\frac{1}{2}$ | 60 X 7 $\frac{1}{2}$ | |

PRINCIPAL DIMENSIONS OF BOILERS.

| General description of Boiler | FIREBOX AND GRATE. | | | | | HEATING-SURFACE. | | | | |
|--------------------------------------------------------------|--------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|
| | Loco. | Loco. | Vertical | Loco. | Loco. | Loco. | Loco. | Loco. | Loco. | Loco. |
| Area of ordinary grate in sq. feet | 6.68 | 3.35 | 2.6 | 6.7 | 5.73 | 5.93 | 3.67 | 3.35 | 5.33 | 6.7 |
| Area of grate as used at trial in sq. feet | 5.98 | 2.63 | 2.6 | 3.39 | 4.69 | 4.18 | 3.67 | 2.63 | 4.32 | 3.39 |
| Width of bar in inches | | | | | | | | | | |
| Width of air-spaces in inches | | | | | | | | | | |
| Area of air-spaces in sq. feet | 2.2 | 0.6 | 1.6 | 1.08 | 1.30 | 2.17 | 1.30 | 0.96 | 1.24 | 1.08 |
| Area of air-spaces as used at trial in sq. feet | 1.7 | 0.77 | 1.51 | 1.11 | 1.16 | 1.53 | 0.75 | 0.77 | 1.1 | 1.11 |
| Height of firebox crown above bars in inches | 36 $\frac{1}{2}$ | 30 $\frac{1}{2}$ | 24 $\frac{1}{2}$ | 30 $\frac{1}{2}$ | 32 $\frac{1}{2}$ | 29 | 31 $\frac{1}{2}$ | 23 $\frac{1}{2}$ | 31 | 30 $\frac{1}{2}$ |
| HEATING-SURFACE. | | | | | | | | | | |
| Length from out to out of tube-plates in inches | 75 | 72 | 35 | 81 $\frac{1}{2}$ | 84 $\frac{1}{2}$ | 70 $\frac{1}{2}$ | 82 $\frac{1}{2}$ | 66 | 84 $\frac{1}{2}$ | 81 $\frac{1}{2}$ |
| Number of tubes | 23 | 24 | 27 | 51 | 53 | 30 | 22 | 76 | 53 | 51 |
| Material of tubes | Iron | Steel | Iron | Iron | Steel | Iron | Iron | Steel | Steel | Iron |
| Outside diameter of tubes in inches | 2 $\frac{1}{2}$ | 1 $\frac{1}{2}$ | 1 $\frac{1}{2}$ | 2 | 1 $\frac{1}{2}$ | 2 $\frac{1}{2}$ | 2 $\frac{1}{2}$ | 1 $\frac{1}{2}$ | 2 | 2 |
| Heating-surface of tubes in sq. feet | 37.2 | 20.45 | 13.8 | 34.5 | 40.8 | 28.3 | 23.0 | 17.0 | 28.1 | 34.5 |
| Heating-surface of tubes in sq. feet | 106.9 | 186.5 | 35.4 | 180.1 | 194.2 | 135.6 | 98.3 | 172.6 | 194.2 | 186.1 |
| Heating-surface of smokebox in sq. feet | 4.59 | 2.6 | | 3.5 | 3.1 | 3.8 | 3.6 | 2.6 | 4.4 | 3.5 |
| Total water-heating surface in sq. feet | 148.7 | 211.5 | 49.0 | 218.1 | 238.1 | 167.7 | 124.9 | 192.2 | 226.7 | 218.1 |
| Sq. feet of water-heating surface to nominal H. P. of engine | 18.6 | 26.4 | 16.3 | 27.2 | 29.6 | 20.9 | 15.6 | 24.0 | 28.3 | 27.2 |
| Ratio of area through tubes to normal grate-area | .122 | .274 | .127 | .092 | .164 | .171 | .171 | .274 | .176 | .092 |

* This includes 8.37 square feet of heating-surface, due to the eight Paxman water-tubes in the firebox.

PRINCIPAL DIMENSIONS OF BOILERS.—Continued.

| Catalogue number..... | SIMPLE ENGINES. | | | | | COMPOUND ENGINES. | | | | | |
|--------------------------------------------------------------|---------------------|-------------------------------------------------|---------------------|---------------------|---------------------|---------------------|---------------------|-------------------------------------------------|---------------------|---------------------|---------------------|
| | 3411 | 3114 | 147 | 3108 | 3125 | 3117 | 3115 | 3113 | 3124 | 3107 | 3116 |
| Area of chimney in sq. feet | .505 | .49 | .248 | .57 | .49 | .6 | .44 | .586 | .49 | .57 | .6 |
| Area of blast-nozzle in sq. inches | 2.4 | 3.7 | .785 | 3.14 | 1.76 | 4.9 | 2.07 | 3.14 | 1.76 | 3.14 | 4.9 |
| Ratio of heating-surface to normal grate-area | 22.2 | 63.1 | 18.8 | 32.5 | 41.5 | 28.1 | 34.0 | 57.3 | 45.5 | 32.5 | 28.1 |
| Weight of water in boiler when quite full in lbs. | 2436 | 1032 | 728 | 7352 | 2451 | 2573 | 2212 | 1687 | 2598 | 2415 | 2492 |
| Weight of water at normal level in lbs. | 1680 | 1124 | 329 | 1428 | 1317 | 1477 | 1346 | 987 | 1393 | 1448 | 1344 |
| Steam-space in cubic feet | 12.05 | 12.83 | 6.41 | 14.85 | 18.23 | 17.62 | 12.49 | 11.3 | 19.37 | 15.86 | 18.45 |
| Height of bottom of gauge-glass above firebox in inches..... | Level | 1½ | 8½ | 2½ | 3½ | 1½ | 2½ | 1½ | 3½ | 2½ | 1½ |
| Arrangement of ashpan damper. | Hinged ashpan-door. | No damper to ashpan, throttle-valve in chimney. | Hinged ashpan-door. | Hinged ashpan-door. | Hinged ashpan-door. | Hinged ashpan-door. | Hinged ashpan-door. | No damper to ashpan, throttle-valve in chimney. | Hinged ashpan-door. | Hinged ashpan-door. | Hinged ashpan-door. |

RESULTS OF EXPERIMENTS CONNECTED WITH THE BOILERS.

(Theoretical evaporative power of the coal from and at 212° = 15.45 lbs. Air theoretically required per 1 lb. of coal 11.38 lbs.)

| | | | | | | | | | | | |
|------------------------------------------------------------------------------------------------|------|--------|--------|--------|--------|--------|-------|-------|--------|--------|--|
| 1. Water supplied to engines.....lbs. | 3182 | 1413 | 667.5 | 1868 | 1674 | 2894 | 2414 | 1394 | 1658 | 1967 | |
| 2. Temperature of water supplied.....F.° | 62 | 63 | 62 | 63 | 63 | 62 | 63 | 65 | 64 | 63 | |
| 3. Average temperature to which the feed-water was raised by condensed steam.....F.° | | 83.6 | | 91.6 | 123.6 | 104.5 | 154.3 | 82.5 | 118.2 | 91.4 | |
| 4. Calculated weight of steam condensed in heating feed-water.....lbs. | | 26.6 | | 49.12 | 96.1 | 114.5 | 215 | 22.2 | 84.7 | 51.4 | |
| 5. Calculated weight of steam condensed in jackets from and at 212°.....lbs. | | 161.7 | | 272.5 | 230.5 | 214.8 | | 215 | 269.8 | 280.4 | |
| 6. Coal consumed.....lbs. | 494. | 138.25 | 113 | 199.75 | 193 | 351 | 259 | 148.5 | 168 | 202.5 | |
| 7. Water evaporated from and at 212° exclusive of jacket-water.....lbs. | 3676 | 1630.8 | 771.09 | 2156 | 1032 | 3343 | 2786 | 1606 | 1912 | 2270 | |
| 8. Water as above evaporated per 1 lb. of coal..... | 9.1 | 11.79 | 6.82 | 10.79 | 10.01 | 9.52 | 10.75 | 10.81 | 11.38 | 11.21 | |
| 9. Estimated weight of water evaporated, including that from jackets from and at 212°.....lbs. | | 1792.5 | | 2428 | 2162.5 | 3557.8 | | 1821 | 2181.7 | 2550.4 | |
| 10. Water as above evaporated per 1 lb. coal from and at 212°.....lbs. | | 12.96 | | 12.27 | 11.21 | 10.14 | | 12.26 | 12.99 | 12.59 | |

* This includes 8.37 square feet of heating-surface, due to the eight Paxman water-tubes in the firebox x.

RESULTS OF EXPERIMENTS CONNECTED WITH THE BOILERS.—Continued.

| Catalogue number..... | SIMPLE ENGINES. | | | | | | COMPOUND ENGINES. | | | | |
|---------------------------------------------------------------------------------------------------|-----------------|-------|-------|-------|-------|--------|-------------------|-------|-------|-------|------|
| | 3111 | 3114 | 147 | 3108 | 3125 | 3117 | 3115 | 3113 | 3124 | 3107 | 3116 |
| 11. Efficiency of boiler and of its management, taking the theoretical value of the coal at 15.45 | .589 | .839 | .442 | .786 | .725 | .656 | .696 | .794 | .840 | .814 | |
| 12. Weight of air passed through per 1 lb. of coal burnt..... lbs. | 23.17 | 12.42 | 16.91 | 26.20 | 23.45 | 17.435 | 19.03 | 15.22 | 24.43 | 27.43 | |
| 13. Percentage of air in relation to theoretical quantity 11.38 lbs. per 1 lb..... % | 203.6 | 109.1 | 148.6 | 230.3 | 206.1 | 153.2 | 167.2 | 132.7 | 214.7 | 241.0 | |

RESULTS OF EXPERIMENTS RELATING TO THE HORSE-POWER AND THE COAL-CONSUMPTION OF THE ENGINES.

| | | | | | | | | | | | | |
|--------------------------------------------------------------------------------------------------------|--------|--|--|--|--|--|--|--|--|--|--|--------|
| 1. Number of brake used..... | 3 | | | | | | | | | | | 3 |
| 2. Time running..... minutes | 245 | | | | | | | | | | | 146 |
| 3. Number of revolutions.....total | 37,342 | | | | | | | | | | | 26,549 |
| 4. Revolutions per minute..... | 152.4 | | | | | | | | | | | 181.8 |
| 5. Declared number of revolutions per minute..... | 150 | | | | | | | | | | | 185 |
| 6. Total weight hanging on brakes..... lbs. | 187.75 | | | | | | | | | | | 265.75 |
| 7. Circumference of brake-load circle.....feet | 17.32 | | | | | | | | | | | 17.31 |
| 8. Coal consumed during runs..... lbs. | 404 | | | | | | | | | | | 202.5 |
| 9. Coal consumed per hour..... lbs. | 98.93 | | | | | | | | | | | 102.3 |
| 10. Indicated horse-power..... | 16 | | | | | | | | | | | 22.1 |
| 11. Declared horse-power..... | 15.02 | | | | | | | | | | | 20 |
| 12. Brake horse-power, excluding correction for friction of brakes..... | 15.31 | | | | | | | | | | | 20.77 |
| 13. Brake horse-power, with friction of brakes added..... | 6.588 | | | | | | | | | | | 21.07 |
| 14. Coal consumed per hour per brake horse-power, excluding friction of brakes..... lbs. | 6.466 | | | | | | | | | | | 2.267 |
| 15. Coal consumed per hour per brake horse-power, with friction of brakes added..... lbs. | 16.19 | | | | | | | | | | | 1.850 |
| 16. Horse-power as calculated at the time of the trials..... | 6.11 | | | | | | | | | | | 22.02 |
| 17. Coal consumed per brake horse-power as calculated at trials, and published in the papers..... lbs. | | | | | | | | | | | | 2.09 |
| 18. Ratio of indicated horse-power to brake horse-power, with friction allowance, as line 13.. | | | | | | | | | | | | .877 |

* The indicator-pipes supplied to this engine by the maker were very small.

COMPARISON OF COAL CONSUMPTION BETWEEN THE CARDIFF AND NEWCASTLE TRIALS.

| | SIMPLE ENGINES. | | | | | | COMPOUND ENGINES. | |
|---------------------------------------------------------------------------------------------------------------|---------------------------------------------|-------|-------|-------------------|-------|-------|---------------------------------------------|-------------|
| | Newcastle Trials. | | | Cardiff Trials. | | | Newcastle Trials. | |
| | 3125 | 3108 | 3114 | 4942 | 2927 | 4942 | 3124 | 3113 3107 |
| Catalogue Number..... | | | | | | | | |
| Coal consumed per brake horse-power per hour, without allowance for friction of brakes..... | Powell's Duffryn Coal. | | | Llangennech Coal. | | | Powell's Duffryn Coal. | |
| | 2.638 | 2.727 | 2.842 | 2.79 | 2.881 | 2.884 | 1.874 | 1.977 2.267 |
| | Powell's Duffryn reduced to Llangennech. | | | | | | Powell's Duffryn reduced to Llangennech. | |
| Consumption of coal as above, corrected for superior quality of Powell's Duffryn used at Newcastle..... | 2.678 | 2.768 | 2.884 | 2.79 | 2.881 | 2.884 | 1.902 | 2.007 2.301 |
| | Powell's Duffryn Coal. | | | | | | Powell's Duffryn Coal. | |
| Coal consumed per brake horse-power per hour, with allowance for friction of brakes..... | 2.599 | 2.684 | 2.766 | | | | 1.850 | 1.943 2.184 |
| | | | | | | | | |

97. The General Conclusions deducible from this remarkably full and accurate collection of data are correspondingly interesting and important. Classifying the data, computing the quantities of heat, and constructing a balance-sheet, the following table was obtained as illustrating the operation of the prize engine No. 3125 :

| DR. | | CR. | |
|--------------------------------------------------------------------------------------------------------------------------------------|-----------|-----------|-------------|
| | Units. | Units. | Percentage. |
| To the heat developed in furnace— | | | |
| In the combustion of wood: | | | |
| From Carbon..... | 79,331 | 9,557 | .32 |
| “ Hydrogen..... | 4,816 | 3,884 | .13 |
| In the combustion of coal: | | | |
| From Carbon..... | 2,481,400 | 8,374 | .29 |
| “ Hydrogen..... | 337,475 | 129,321 | 4.44 |
| “ Sulphur..... | 9,071 | | |
| | | 53,394 | 1.83 |
| | | 130,980 | 6.34 |
| | | 53,509 | |
| | | 2,090,300 | 71.78 |
| | | 271,307 | 9.32 |
| | | 53,915 | 1.85 |
| | | 107,552 | 3.70 |
| | | 2,912,093 | 100.00 |
| By heat expended: | | | |
| 1. In evaporating the water in the wood and heating its steam to 385° | | | |
| 2. In heating the wood and the air required for its combustion from 70° to 385° | | | |
| 3. In evaporating the water in the coal and heating its steam to 385° | | | |
| 4. In heating the coal and the air required for its combustion from 70° to 385° | | | |
| 5. In displacing the atmosphere by the products of combustion of the wood and the coal with the air needed for their combustion..... | | | |
| 6. In heating the excess of air..... | | | |
| In displacing the atmosphere by the excess of air..... | | | |
| 7. In evaporating the water in the boiler. | | | |
| 8. In radiation and convection..... | | | |
| 9. In ashes and unconsumed fuel..... | | | |
| 10. Unaccounted for..... | | | |
| | 2,912,093 | | |

It will be seen that the prize simple engine at Newcastle consumed 2.678 lbs. of coal per brake horse-power, while the prize compound engine consumed only 1.902 lbs. of coal per brake horse-power, a gain of 32 per cent., or nearly one-third.

In the preceding table, the engine No. 2927, tested at Cardiff, and the two prize engines and compound engine No. 3113, tested at Newcastle, are compared. The third line gives the absolute temperature of the steam in each case; the fourth line the fall of temperature, on the supposition that the steam leaves the cylinder at a temperature proper to 1 lb. back-pressure, that is, 215°; the fifth line is the quotient of the division of the fourth by the third, and shows the proportion of work to be expected; the sixth line is the reciprocal of the fifth reduced to one as the standard of engine No. 2927, and represents the proportion in which the steam should have been consumed, that being of course inversely as the amount of duty to be expected.

We see that simple engine No. 3125 should have demanded about 7 per cent. less steam, and compound engine No. 3124 23½ per cent. less than engine No. 2927. In reality the simple engine, as will be seen by the eighth line, took 13 per cent. less, while the compound took nearly 30 per cent. less than the engine No. 2927.

Considered apart from their boilers, it will appear that simple engine No. 3125 and compound engine No. 3124 each in round numbers exceeded by some 6 per cent. the duty which, having regard to the increase of the pressure of steam above that of the engine tried at Cardiff, and taken here as a standard, would have led one to expect, while engine No. 3113 used an amount of steam which was about 6 per cent. more than the foregoing calculation anticipated; so that the result fell a little below that of compound No. 3124, working at 100 lbs. less pressure. These trials appear to point to the conclusion that, with our present state of knowledge, it is probable that pressures between 150 and 200 lbs. per square inch will give the best practical results with compound engines of these types.

The last column in the balance-sheet shows the percentage

which each source of loss bears to the total amount of heat generated. Heating the fuel, and the air necessary for its combustion, and displacing the atmosphere (items 4 and 5) take $6\frac{1}{4}$ per cent.; while the cost of dealing with the excess air amounts to $6\frac{1}{8}$ per cent. The loss by cooling is, however, the most serious of all, and, although this engine was, as regards the usual parts, lagged with exceptional care, amounted to $9\frac{1}{8}$ per cent. The losses which cannot be certainly accounted for amount to less than $3\frac{3}{4}$ per cent. A portion of these were probably due to the increased rate of cooling while the engine was at work, for the cylinders, piston-rods, valve, spindles, and the working-parts generally were hotter, and therefore emitted more heat, than when at rest.

Reverting to the balance-sheet and to its credit side, it will be seen that items 1 to 5, involving an absorption of 7 per cent. of the heat produced by the fuel, are inevitable losses; but item 6, which relates to the excess of air, and comprises the two heads of heating that excess, and of displacing the atmos-

COMPARISON BETWEEN THE THEORETICAL AND ACTUAL ECONOMY DERIVED FROM AN INCREASE OF STEAM-PRESSURE.

| | CARDIFF. | NEWCASTLE-ON-TYNE. | | |
|-----------------------------------------------------------------------------------------------------------------------------------------------------|----------|--------------------|-----------|-------|
| | Simple. | Simple. | Compound. | |
| Catalogue Number..... | 2927 | 3125 | 3124 | 3113 |
| 1. Steam-pressures above atmosphere, lbs. | 80 | 95 | 150 | 250 |
| 2. Temperature of steam.....F.° | 324 | 334 | 365 | 406 |
| 3. Corresponding absolute temperatures.....F.° | 784 | 794 | 825 | 866 |
| 4. Falls of temperature to 215° or 675° absolute.....F.° | 109 | 119 | 150 | 191 |
| 5. Proportions which the falls bear to the original absolute temperatures. | .139 | .150 | .182 | .220 |
| 6. The reciprocals of the above ratios, to which reciprocals the fuel actually consumed should correspond, reduced to engine No. 2927 as unity..... | 1 | .927 | .763 | .632 |
| 7. Water actually consumed per brake horse-power per hour (not including jacket-water).....lb. | 30.22 | 26.40 | 21.33 | 21.38 |
| 8. Relative proportion of water used... | 1 | .873 | .706 | .707 |

phere for its reception, gives a further amount of loss = 6.34 per cent., which is preventable. In the case of this engine, No. 3125, we have an excess of air weighing 12.31 lbs. for each lb. of fuel burnt, being practically equal to the air which is theoretically needed,—while in engine No. 3114 the excess was only 1.67 lbs., and in engine No. 3113, 4.02 lbs. It is clear, therefore, that if 3125 had been worked with no greater excess than 3114, the 6.34 per cent. of loss of item 6 would have been reduced by 5.49 per cent., leaving only .85 per cent.

*Analysis by Messrs. Pattinson and Stead, Middlesbrough, of
Powell-Duffryn's Coal used at the Newcastle Trials.*

Samples were collected at intervals during the trials, and the coal analyzed was an average of these:

| | | |
|-------------------|--------------|--------------------------------|
| Carbon, . . . | 88.40 | available |
| Hydrogen, . . | 3.65 — 0.32 | = 3.33H |
| Oxygen, . . . | 2.55 = 0.32H | = 2.87H ₂ O (water) |
| Nitrogen, . . . | 0.64 | |
| Sulphur, . . . | 0.76 = 1.36 | per cent. pyrites |
| Ash, | 3.17 | |
| Water, | 0.83 | |
| | <hr/> | |
| | 100.00 | |
| | <hr/> | |
| Sulphur in ash, . | 0.04 | |

Calorific value in British Thermal Units.

| | | | | | | | |
|--------------------------------|-------|---|--------|-------|---|--------|-------|
| Carbon, . . . | .884 | × | 14,544 | units | = | 12,856 | units |
| Hydrogen, . . | .0333 | × | 61,200 | " | = | 2,037 | " |
| Pyrites estimated at | | | | | | 47 | " |

Total per one pound of coal, . . 14,930 units

Weight of air required to burn one pound of coal, 11.38 lbs.

98. Trials of Locomotive Engines are more difficult of prosecution than are those of any other class of steam-engine. The conditions of its operation are such that it is very difficult to secure measurements of coal and of water consump-

tion, exceedingly difficult to obtain a good arrangement for taking indicator-diagrams, and next to impossible to determine the quality of steam made while in regular work. It is commonly expected that the engineer conducting the trials will report on the following points :

(1) The dimensions of the engines, as to diameter of cylinder, stroke, diameter of boilers, exhaust-nozzles, etc.

(2) Their weight and the distribution upon driving-wheels.

(3) The weight of the train hauled.

(4) The weight of coal consumed in hauling the same train over the same route by each engine.

(5) The quantity of water evaporated by each engine in doing the work.

(6) The relative amounts of smoke and cinders which escaped from the smoke-stacks.

(7) The temperature of the gases escaping.

(8) The tractive resistance of the train at the same places as indicated by a dynamometer during the trips of each engine.

(9) The pressure of steam in the cylinders of the several engines, as shown by indicator-diagrams, the pressure indicated in the boilers being recorded at the same time.

(10) The time occupied in making each trip and between points, which should be as nearly uniform as possible.

(11) The temperature of the air at starting, in the middle, and at the end of each trip.

(12) The temperature of the water in the tenders.

Such trials should usually be made as nearly as possible under the ordinary conditions of every-day work.

The engines are weighed in working trim, with steam up and ready to start, and also with steam off and water blown out. The coal is weighed by taking the weight of tender empty and loaded, and noting the difference in weight as that of coal consumed on the run, returning the balance at its end. The water is commonly measured by using a float in the tank, the rise and fall of which, and the area of water-surface, being known, the volume and weight of water become easily determinable. The water may also be weighed, as well as the coal. The rela-

tive quantities of smoke and cinder ejected can only be estimated from observation. The engine should be given a preliminary provisional trial to see that everything is in working order.

The following is an abstract of a good example of a report describing work of this kind under the direction of Mr. Leavitt, the trial being conducted by Professor Coon :

The object of these tests was to determine the relative efficiency of Strong's locomotive boiler and cylinder and valve gear, and their ease of working and liability to derangement, as compared with the best type of American locomotive in common practice at the present time. To this end three locomotives were tested, viz.: Engine No. 444, fitted with Strong's twin-furnace boiler and a four-valve cylinder and valve gear; engine No. 383, having an ordinary straight-top boiler, with fire-box over instead of between the frames, anthracite-coal grates, and fitted with cylinder and valve gear similar to No. 444; engine No. 357, having an ordinary boiler similar to that on engine No. 383, save that it has a "wagon top" of eight inches, and the link-motion common in American practice, with plain slide-valves having a good balancing device.

The leading particulars of the three locomotives are as follows :

| | ENGINE 444. | ENGINE 383. | ENGINE 357. |
|--------------------------------------------------------------------|------------------------------------------------|--------------------------------|------------------------------------------|
| CYLINDERS, ETC. | | | |
| Cylinder, diameter and stroke..... | 20 ins. by 24 ins.. | 19 ins. by 24 ins | 20 $\frac{1}{2}$ by 24 ins. |
| Diameter of piston-rod..... | 3 $\frac{1}{2}$ ins..... | 3 $\frac{1}{2}$ ins | 2 $\frac{3}{8}$ and 2 $\frac{3}{4}$ ins. |
| Length of connecting-rod, centres ... | 8 ft. 3 ins | | |
| Transverse distance between cylinder, centres | 7 ft..... | | |
| Distance from centre of main drivers to centres of cylinders | 12 ft. 10 ins..... | | |
| Number of valves per cylinder..... | 4, 2 steam, 2 exh.. | 4, 2 steam, 2 exh.. | One. |
| Type of valves..... | Gridiron..... | Gridiron | Balanced slide. |
| Number of ports per valve..... | 10 | 10 | |
| Size of ports..... | 4 $\frac{1}{8}$ ins. by $\frac{3}{4}$ in | | |
| Full travel of valves..... | 1 $\frac{1}{8}$ in | 1 $\frac{1}{8}$ in | |
| Lap of valves | $\frac{1}{8}$ in | | |
| Lead of valves, steam | $\frac{1}{8}$ in. constant.... | $\frac{1}{8}$ in. constant.... | |
| Lead of valves, exhaust | $\frac{1}{8}$ in. constant.... | $\frac{1}{8}$ in. constant.... | |
| Throw of eccentrics | 2 $\frac{1}{2}$ ins | | |
| Tractive force per lb. of mean effective pressure on piston | 154.8 pounds... .. | 131.3 pounds..... | 149.11 pounds. |
| Cylinder clearance in cubic inches... .. | 481 | 448 | 568. |
| Cylinder clearance in per cent. of piston displacement..... | 6.38 per cent..... | 6.58 per cent..... | 7.35 per cent. |

| | ENGINE 444. | ENGINE 383. | ENGINE 357. |
|-----------------------------------------------------------------|--------------------------------|---------------------|-----------------|
| WHEELS AND JOURNALS. | | | |
| Driving and truck wheel centres (all tires of steel)..... | Wrought-iron.... | Cast-iron..... | Cast-iron. |
| Front truck 4-wheeled swing beam... Yes..... | Yes..... | Yes..... | Yes. |
| Rear truck 2-wheeled swing and radius bar..... | Yes..... | None..... | None. |
| Nominal diameter of driving-wheels... 62 ins..... | 62 ins..... | 66 ins..... | 68 ins. |
| Caliper diameter of driving-wheels... 62½ ins..... | 62½ ins..... | 65½ ins..... | 66½ ins. |
| Diameter of front truck-wheels..... 30 ins..... | 30 ins..... | 30 ins..... | 30 ins. |
| Diameter of rear truck-wheels..... 42 ins..... | 42 ins..... | 42 ins..... | 42 ins. |
| Total wheel-base of engine..... 30 ft. 2 ins..... | 30 ft. 2 ins..... | 22 ft. 9 in..... | 22 ft. 1 in. |
| Rigid wheel-base of engine..... 5 ft. 7 ins..... | 5 ft. 7 ins..... | 7 ft. 9 in..... | 7 ft. |
| Driving axle-journals (diameter and length)..... | 7½ in. by 10 ins... | | |
| Front truck axle journals (diameter and length)..... | 6 ins. by 11 ins... | | |
| Rear truck axle journals (diameter and length)..... | 7 ins. by 9 ins... | | |
| Main crank-pin journals (diameter and length)..... | 5 ins. by 6 ins... | | |
| Coupling-rod journals (diameter and length)..... | 4½ ins. by 4 ins... | | |
| WEIGHTS, ETC. | | | |
| Weight on first pair drivers, in working order..... | 30,000 pounds.... | | |
| Weight on second pair drivers, in working order..... | 30,000 "..... | | |
| Weight on third pair drivers, in working order..... | 30,000 "..... | None..... | None. |
| Total on drivers, in working order... 90,000 "..... | 90,000 "..... | 74,640 pounds... | 63,280 pounds. |
| Weight on front truck, in working order..... | 27,000 "..... | 24,880 "..... | 27,440 " |
| Weight on rear truck, in working order... 21,000 "..... | 21,000 "..... | None..... | None. |
| Total weight of engine, in working order..... | 138,000 "..... | 99,520 pounds... | 90,720 pounds. |
| BOILERS, ETC. | | | |
| Height of boiler-centre above rail.... 7 ft. 3 ins..... | 7 ft. 3 ins..... | | |
| Kind of boiler..... | Twin-furnace.... | Straight-shell.... | Wagon-top. |
| Material for boiler-plate..... | O. H. steel..... | O. H. steel..... | O. H. steel. |
| Diameter of barrel inside smallest ring 58 ins..... | 58 ins..... | 55 ins..... | 54 ins. |
| Diameter of fire-boxes and combustion chambers, corrugated..... | 38½ ins. inside..... | | |
| Length of grates..... | 42½ ins. outside... 11 ft..... | 11 ft..... | 11 ft. |
| Width of grates..... | 3 ft. 4½ ins..... | 3 ft. 4½ ins..... | 3 ft. 6½ ins. |
| Number of tubes, all iron..... | 306..... | 229..... | 248. |
| Diameter of tubes, outside..... | 1½ ins..... | 2 ins..... | 2 ins. |
| Length of tubes..... | 11 ft. 5 ins..... | 11 ft. 4½ ins..... | 12 ft. 2 ins. |
| Grate area, square feet..... | 62 square ft..... | 37.12 square ft.... | 39.2 square ft. |
| Heating-surface, fire-box, square feet... 155..... | 155..... | 151.6..... | 142.3. |
| Heating-surface, combustion chamber, square feet..... | 93..... | | |
| Heating-surface, tubes, square feet.... 1600..... | 1600..... | 1234.3..... | 1429.8. |
| Heating-surface, total, square feet.... 1848..... | 1848..... | 1385.9..... | 1572.1. |
| Ratio of heating-surface to grate area... 29.8 to 1..... | 29.8 to 1..... | 37.3 to 1..... | 40.1 to 1. |
| Smallest inside diameter of smoke-stack..... | 16 ins..... | | |
| Height of top of smoke-stack above rail..... | 14 ft. 3 ins..... | | |
| Working steam-pressure per sq. inch... 160 lbs..... | 160 lbs..... | 160 lbs..... | 140 pounds. |
| TENDER. | | | |
| Eight-wheeled, double trucks (diameter of wheels)..... | 33 ins..... | 33 ins..... | 33 ins. |
| Capacity of tender (gallons)..... | 3000..... | 3000..... | 3000 |
| Capacity of tender (coal)..... | 10,000 pounds.... | 10,000 pounds.... | 10,000 pounds. |
| Weight of tender loaded..... | 70,000 "..... | 70,000 "..... | 70,000 " |
| (Same tender used on all tests.) | | | |

All the locomotives were subjected to exactly the same work, under similar conditions. The route was a continuous succession of curves as sharp as 14 degrees, and grades as steep as 96 feet per mile.

The water supplied to the tender on each trip was gauged as follows: On each side of the tender, and at the centre of gravity of the water space, long glass gauges, the height of the tender, were attached, with a blank wooden scale behind. The tender was filled nearly full of water and placed on the track scales, the height of water in the glass gauges being marked on the scale. These were the 0 marks. The water was then drawn off five cwt. at a time (560 pounds), and corresponding marks made on the scale, each division thus representing 560 pounds of water. The readings of both scales were taken immediately before and after taking water, and at the end of the run. The same tender (that of engine No. 444) was used on all the trials, without alteration.

All the coal for each run was weighed in barrows on a platform scale, and any coal remaining in the tender at the end of the run was deducted from the original amount, it also being weighed. At the beginning of each run, before any coal was charged to the experiment, the engine was allowed to start with a uniformly good fire. It was not practicable to get the weight of the ashes.

The boiler-pressure was taken at five-minute intervals on each run.

The temperature of the feed-water at the three tanks where water was taken did not vary half a degree from 64.3° F. throughout the trials.

Engine No. 357 has a boiler feed-pump attached to one cross-head, any deficiency of water being supplied by an injector. Engines Nos. 444 and 383 were supplied with water solely by injectors.

All indicator-cards were taken on up-grade. Two observers took simultaneous cards from each end of both cylinders, while a third, in the cab, noted the steam-pressure, position of throttle-lever and reverse-lever, the exact time the cards were taken,

and the exact time of passing the mile-posts, and the time of stopping and starting at stations.

The indicator-rig, for reducing the piston-motion for the indicator-barrel, was a modified true pantagraph motion. The strings used were hard-braided linen, about 8 inches long. Cards taken at 60 miles per hour are not more than .03 inch longer than those taken at one mile per hour. No wind-shields were used, and not the slightest difficulty was experienced in taking accurate cards at over 60 miles per hour.

Through the courtesy of the Pennsylvania Railroad, their dynamometer car was used.

Attention is invited to the tables of coal consumed and water evaporated. On the first four trips with Engine 444 some journals on the front truck heated, and on the third trip a main crank-pin heated, so that a helper was considered desirable. Subsequently no parts warmed at all on any of the engines.

If the consumption of coal of the three locomotives be compared, each on two trips when they were using the same grade of coal, to wit: Locomotive No. 444, on trips No. 4 and No. 10; locomotive No. 383, on trips No. 8 and No. 14; and locomotive No. 357, on trips No. 12 and No. 13,—it gives the average for the three respectively, as follows:

| | | |
|---------------------------------|-------|-------------|
| Average for engine No. 444, . . | 6,537 | pounds coal |
| “ “ “ “ 383, . . | 7,441 | “ |
| “ “ “ “ 357, . . | 8,087 | “ |

—which is a difference of 646 pounds between Nos. 383 and 357, or an advantage of 8.7 per cent. in favor of engine No. 383, and a difference of 1550 pounds between Nos. 444 and 357, or an advantage of 23.7 per cent. in favor of No. 444. It is also to be borne in mind that engine No. 357 has 2 square feet more grate area and nearly 200 square feet more heating-surface and much better steam room than No. 383. With equal boilers there would be still greater difference in the coal. In support of this, compare the two runs of engine No. 383 on May 2 and May 10, and engine No. 357 on May 7 and May 9,

as to water required from Mauch Chunk to Glen Summit, all the way up grade, and no blowing-off took place. On these four runs the load was precisely the same, about 421600 pounds, besides engine and tender. The mean consumption of water from Mauch Chunk to Glen Summit for the two runs mentioned, for the two engines, was—

For No. 383, 17,942 pounds

“ 357, 21,585 “

—a difference of 3643 pounds in favor of No. 383, or 20.3 per cent.

COAL-CONSUMPTION AND WATER-EVAPORATION.

| DATE. | TRIP. | KIND OF COAL. | POUNDS COAL CONSUMED. | POUNDS WATER EVAPORATED. | POUNDS OF WATER PER POUND COAL. |
|------------------------|-------|----------------------------------------------------------------------------|-----------------------|--------------------------|---------------------------------|
| <i>Locomotive 444.</i> | | | | | |
| Apr. 25 | 1st | Anthracite, mixed sizes; mostly screenings; dirty..... | 8,114 | 51,940 | 6.40 |
| “ 26 | 2d | Anthracite, mixed sizes; mostly screenings; dirty..... | 7,698 | 52,132 | 6.77 |
| “ 27 | 3d | Anthracite, egg size; clean; had hot pin and helper.... | 6,386 | Not taken | ... |
| “ 28 | 4th | Anthracite, egg size; clean..... | 6,562 | 51,900 | 7.91 |
| “ 29 | 5th | Bituminous, “Blacksmith's” wet; had helper..... | 7,216 | 53,816 | 7.45 |
| May 4 | 9th | Anthracite, pea size; clean..... | 8,002 | 50,680 | 6.33 |
| “ 5 | 10th | Anthracite, egg size; dirty..... | 6,511 | 50,708 | 7.78 |
| “ 6 | 11th | Bituminous, lump size; Westmoreland Company's Mines; had helper twice..... | 5,614 | 48,062 | 8.56 |
| “ 11 | 15th | Anthracite, lump size; Hillman Mine..... | 6,948 | 51,422 | 7.40 |
| “ 18 | 16th | Bituminous, Barclay; broken sizes..... | 7,530 | 53,970 | 7.16 |
| “ 19 | 17th | Bituminous, Barclay; broken sizes..... | 7,154 | 52,234 | 7.30 |
| <i>Locomotive 383.</i> | | | | | |
| Apr. 30 | 6th | Anthracite, lump; Hillman Mine..... | 8,643 | 45,472 | 5.26 |
| May 2 | 7th | Anthracite, lump; Hillman Mine..... | 8,311 | 43,512 | 5.23 |
| “ 3 | 8th | Anthracite, lump; Franklin Mine..... | 7,666 | 42,364 | 5.53 |
| “ 10 | 14th | Anthracite, lump; Franklin Mine..... | 7,217 | 41,734 | 5.78 |
| <i>Locomotive 357.</i> | | | | | |
| May 7 | 12th | Anthracite, lump; Franklin Mine..... | 8,260 | 47,824 | 5.79 |
| “ 9 | 13th | Anthracite, lump; Franklin Mine..... | 7,914 | 49,112 | 6.20 |

The Dynagraph-car Diagrams.

The results will be given of diagrams taken in the Pennsylvania Company's dynamometer car from Sugar Notch to Fairview on May 19 and 20, and from Mauch Chunk to Glen Summit on May 19. Wherever the word “ton” is used it will mean 2240 pounds.

Weight of engine (No. 444) and tender:

| | Tons. |
|---------------------------------------|-------|
| Sugar Notch to Fairview, | 87 |
| Mauch Chunk to Glen Summit, | 85 |

Weight of coaches (including dynagraph car):

| | |
|-----------------------------------------------|-----|
| Sugar Notch to Fairview, May 19, | 125 |
| " " " " 20, | 174 |
| Mauch Chunk to Glen Summit, May 19, | 200 |

Per cent. which weight of train is of weight of engine and tender:

| | |
|-----------------------------------------------|------|
| Sugar Notch to Fairview May 19, | 69.6 |
| " " " " 20, | 50.0 |
| Mauch Chunk to Glen Summit, May 19, | 42.5 |

Traction of train only, as given by the dynamometer-diagrams (mean):

| | Pounds. |
|-------------------------------------------------------|---------|
| Sugar Notch to Fairview, May 19, | 6,208 |
| " " " " 20, | 7,930 |
| Mauch Chunk to Penn Haven Junction, May 19, | 5,381 |
| Penn Haven Junction to White Haven, May 19, | 4,635 |
| White Haven to Glen Summit, May 19, | 7,540 |

Tractive force due to indicated horse-power:

| | Pounds. |
|-------------------------------------------------------|---------|
| Sugar Notch to Fairview, May 19, | 11,198 |
| " " " " 20, | 12,220 |
| Mauch Chunk to Penn Haven Junction, May 19, | 8,813 |
| Penn Haven Junction to White Haven, May 19, | 7,549 |
| White Haven to Glen Summit, May 19, | 10,945 |

Tractive force due to engine and tender (the difference between traction due to indicated horse-power and traction due to train):

| | Pounds. |
|-------------------------------------------------------|---------|
| Sugar Notch to Fairview, May 19, | 4,990 |
| " " " " 20, | 4,290 |
| Mauch Chunk to Penn Haven Junction, May 19, | 3,432 |
| Penn Haven Junction to White Haven, May 19, | 2,914 |
| White Haven to Glen Summit, May 19, | 3,405 |

In the following table, column 1 gives the per cent. which the weight of engine and tender is of the weight of the train, while column 2 gives the per cent. which traction due to engine and tender is of traction due to train :

| | 1. | 2. |
|---------------------------------------------|------|------|
| Sugar Notch to Fairview, May 19, | 69.6 | 80.0 |
| " " " " 20, | 50.0 | 54.0 |
| Mauch Chunk to Penn Haven Junction, May 19, | 42.5 | 63.0 |
| Penn Haven Junction to White Haven, May 19, | 42.5 | 62.9 |
| White Haven to Glen Summit, May 19, . . . | 42.5 | 45.1 |

Strain-diagram A fairly represents the action of this locomotive. The portion shown was taken on a 96-foot grade and on a 10-degree curve, and at a speed of 13.1 miles per hour, when the locomotive was working at its best. Strain-diagram B is a portion of that taken on May 20, with engine No. 444, the portion shown being taken at precisely the same portion of the road as diagram A, viz.: on 96-foot grade and 10-degree curve, but at a speed of 27.2 miles per hour. The revolutions of the drivers of engine No. 82 can be accurately taken from the strain-diagram, every cusp on either side of the curve representing a revolution.

It is obvious from this diagram (A) that one end of one cylinder was doing more work than the other end of the other cylinder. Hence the great oscillations in the diagram. It is also to be noted that the train, as a whole, had its rate of oscillation about every six or seven revolutions of the drivers.

Speed of trains:

| | Miles per Hour. |
|---------------------------|-----------------|
| Engine No. 82, | 13.1 |
| Engine No. 444, | 27.2 |

Net tractive force at time diagrams A and B were taken:

| | Pounds. |
|-------------------------------|---------|
| For engine No. 82, | 15,097 |
| For engine No. 444, | 8,718 |

Resistance or tractive force, in pounds, per long ton :

| | |
|-------------------------------------------|------|
| For engine No. 82 (diagram A), | 53.5 |
| For engine No. 444 (diagram B), | 50.1 |

Cylinders, 20 in. x 26 in.; drivers, 50 in. diameter; weight of train, 281.7 long tons; 96-foot grade; 10-degree curve; speed, 13.1 miles per hour; net horse-power, 527.4; net tractive force, 15,097 pounds; tractive force per ton, 53.5 pounds.

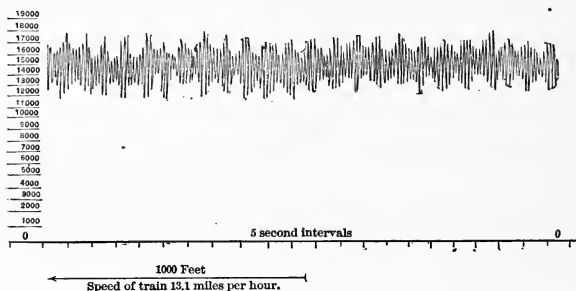


FIG. 122.—STRAIN-DIAGRAM "A," LOCOMOTIVE No. 82.

Weight of train, 147 long tons; 96-foot grade; 10-degree curve; speed, 27.2 miles per hour; indicated horse-power, 952.4; net horse-power, 632.3; net tractive force, 8718 pounds; tractive force per ton, 50.1 pounds.

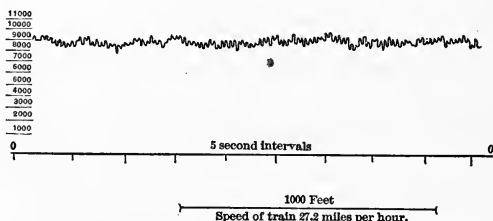


FIG. 123.—STRAIN-DIAGRAM "B," LOCOMOTIVE No. 444.

For the above strain-diagrams, A and B, the net horse-power, due to tractive force of 15,097 pounds and 8718 pounds for the respective speeds are: For engine No. 82, net horse-power, 527.4; for engine No. 444, net horse-power, 632.3; or the net

horse-power of engine No. 82 was 83.4 per cent. of net horse-power of engine No. 444.

The indicated horse-power at the time diagram B was taken is as follows :

Mean effective pressures, in pounds, per square inch :

| | P. R. End. | Front End. |
|---------------------------------------------|---------------|---------------|
| Right-hand cylinder, | 88.64 | 85.84 |
| Left-hand " | 90.21 | 82.48 |
| Total, | <u>178.85</u> | <u>168.32</u> |
| Indicated horse-power, P. R. end, | | 484.3 |
| " " front " | | <u>468.1</u> |
| Total indicated horse-power, | | 952.4 |

Horse-power due to traction of No. 444 engine and tender, $952.4 - 632.3 = 320.1$. Per cent. which traction of No. 444 engine and tender was of traction of train, 50.6 per cent. Per cent. which weight of No. 444 engine and tender was of weight of train, 50.0 per cent.

As the weight of train drawn by engine No. 444 was in part estimated, it is probably not correct within 0.6 per cent. The weight of train drawn by engine No. 82 was obtained exactly.

The following is a *résumé* of experiments on locomotives, made by M. Marié, engineer of the Paris and Lyons Railroad, on the heavy grades leading to the Mont Cenis Tunnel, which may be compared with the preceding :

| | |
|---------------------------------------------------------------------------------------------------------------------|--------------|
| Water evaporated by pound of coal, | 8.88 |
| Consumption of coal per indicated horse-power, | 2.88 |
| " " " effective " | 3.27 |
| Average speed per hour during trials, | 17.04 |
| Efficiency of boiler, | 65 per cent. |
| Efficiency of engine, as compared to " perfect " engine working under the same range of temperatures, | 53 " |

Dimensions of Locomotive.

| | |
|--------------------------------------------------|------------------------------------|
| Cylinders, $21\frac{1}{4} \times 26$ in. stroke. | Drivers, $49\frac{5}{8}$ in. diam. |
| Heating-surface—fire-box, | 104 sq. ft. |
| “ —tubes | 2,045 “ |
| Total, | 2,149 sq. ft. |
| Grate area, | 22.4 “ |
| Weight of engine and train, | 366,474 lbs. |

The coal was of good quality, yielding 14,600 British thermal units when burnt in oxygen. The amount of ash was 6.5 per cent., and the coal contained 1 per cent. of moisture. The average point of cut-off during the experiments was at 19 per cent. of the stroke.

99. The Several Results and Conclusions to be derived from these, as from locomotive-trials generally, are very obvious. The comparatively small quantity evaporated by the fuel is evidence that the necessarily large amount of work demanded of the locomotive boiler is obtained at great sacrifice of efficiency. Much larger figures than those above given are often quoted, as, for example, in the French report referred to above; but it is probably generally the fact that some priming has produced a misleading result.* The second important matter is the consumption of water, an especially serious item in the performance of locomotives. In the trials above detailed, the weight of water used ranged from 24 to 40 pounds per I. H. P. per hour. The latter is a fair result; the former is remarkably good. The coal account gives from 3 to 5 pounds per I. H. P. per hour. Four pounds seems to be a good average amount. Little need be added to what has already been said in reference to the data and the deductions from the dynamometer records. The traction ranged, in the case given, in the neighborhood of 50 pounds per ton, or about 2.2 per cent.; and the power exerted was from 600 to nearly 1000 horse-power.

The results reported from the French trials quoted are re-

* See Clarke's Manual, p. 799.

markably and exceptionally excellent. Later constructions of compound locomotives, only, have rivalled them. It will be noted that the boiler had very high efficiency, 65 per cent., and that the engine had a total efficiency above one-half that of the ideal, perfect, engine operating under similar external conditions.

The general results of comparisons of performance of simple and compound locomotive engines show an evident gain by the latter, to the extent of from 15 to 20 per cent., in fuel and steam consumption; which gain is often partly compensated by a somewhat greater consumption of oil. The saving in coal consumed is often less than that of steam used; the two quantities being, in some reported cases, 15 and 20 per cent. respectively. The pressure of steam adopted in these comparisons is commonly 10 to 12 atmospheres.

100. Trials of Marine Engines are conducted under somewhat more favorable conditions than attend trials of locomotives, but they are also to some extent embarrassed by the peculiar surroundings of the motor apparatus. Some of the most interesting and fruitful investigations ever made have, notwithstanding these difficulties, been made in this direction. The now well-known trials of the U. S. R. M. steamers, designed by Mr. Emery, are examples of such, and their reported performance is here presented.*

The "Rush," the "Dexter," and the "Dallas" are similar as respects the hulls, the screws, and the boilers; but the engines are different: that of the "Rush" being compound; that of the "Dexter," high-pressure condensing; and that of the "Dallas," low-pressure condensing. The vessels are each $129\frac{1}{2}$ feet between perpendiculars at water-line, 23 feet extreme breadth of beam, and 10 feet depth of hold. The draught of water aft is about 8 feet 10 inches. The hulls are of wood. One of the vessels averaged upward of eleven nautical miles per hour for six consecutive hours on her trial trip, and neither of them averaged less than ten knots.

Each vessel has one boiler 11 feet wide on base and 9 feet

* Reports to Navy and Treasury Departments; 1874-5.

high, three furnaces in each boiler, located between water-legs. The products of combustion return through tubes within the shell. The boiler of the "Dallas," designed for low-pressure steam, is 13 feet 9 inches long. The boilers of the two other vessels were designed for high-pressure steam, and are each 12 feet long. The steam-chimney is connected to the boiler by a large tube. The boiler of the "Dallas" has 160 tubes, $3\frac{1}{4}$ inches in diameter and 9 feet 3 inches long. The boilers of the two other vessels have each 158 tubes, $3\frac{1}{4}$ inches in diameter and 9 feet 8 inches long.

The "Rush" is propelled by a compound engine, with vertical cylinders and intermediate receiver, the pistons being separately connected to cranks at right angles. The cylinders are steam-jacketed, felted, and lagged, and are, respectively, 24 and 38 inches in diameter, with 27 inches stroke of piston. The steam is distributed to the high-pressure cylinder by a short slide-valve with cut-off plates sliding on the back. The distribution of steam to the low-pressure cylinder is effected by means of a double-ported slide-valve with lap proportioned to cut off the steam at about half-stroke. The surface condenser contains 900 square feet of condensing-surface. The air-pump is operated from the cross-head of the low-pressure engine. The circulating-pump is of the centrifugal type, operated by a small engine, directly connected. The screw is 8 feet 9 inches in diameter, with mean pitch of $14\frac{1}{2}$ feet. The engine was intended to be operated regularly with a steam-pressure of 80 pounds, but during the trials, hereafter referred to, it was reduced to correspond to the pressure carried on the trial of the "Dexter."

The "Dexter" is of the inverted type, with a single cylinder 26 inches in diameter and 36 inches stroke of piston. The cylinder is not jacketed. Steam is distributed by a short slide-valve, with adjustable cut-off plates. The condenser is located outside the frame, but it and the pumps are duplicates of those in the "Rush." The engine and boiler are designed to be operated with a maximum steam-pressure of 70 pounds.

The "Dallas" is of the inverted type, with a single cylinder

36 inches in diameter, with 30 inches stroke of piston. The cylinder is not steam-jacketed, but is covered with non-conducting composition and lagged. Steam is distributed by a short slide-valve, with adjustable cut-off plates. The surface condenser is located under starboard frames, and has the same condensing surface as those in the other vessels. The air- and circulating-pumps are also substantially the same. The engine and boiler are designed to be operated with a maximum steam-pressure of 40 pounds.

The experiments were made with the vessel secured to the wharf.

The coal was broken on the wharf to proper size (the vessel's bunkers having been closed and sealed) and filled into bags to a certain weight.

The bags were sent on board, as ordered by the engineer on watch, he making record of the number of bags and the time of receipt; a similar record being made by one of the men on the wharf. At the end of the hour the number of bags on the fire was reported and entered in the appropriate column. The ashes were measured into buckets, of which the mean weight was ascertained and tallied as they were hoisted out.

The feed-water was measured before its return to the boiler; for this purpose a tank of boiler-plate was constructed, having a plate dividing it vertically into two equal parts. In the upper edge of the plate was cut a notch eight inches long, by which the height to which each half of the tank could be filled was determined. The mean of the weight of water in the half-tanks was 1129½ pounds, at a temperature of 72° Fahrenheit. In the computations for each experiment, the weight of water is reduced to correspond with mean temperature.

One of the feed-pumps was disconnected from the check-feed valve, and its discharge-pipe led to a receiving-tank placed over the two halves of the measuring-tank, into which this pump forced the water from the hot-well. The receiving-tank had two cocks, one over each half-tank, so that either could be filled from it at will.

The other feed-pump had its suction-pipe detached from the hot-well and connected with the bottoms of the two half-tanks, through a cock on each, so that the contents of either could be drawn out and discharged.

The method of measuring the water was as follows : One side having been filled, the cock above it on the receiving-tank was closed and the other over the empty half opened. When the water in the full one had settled to the edge of the notch, its cock in the feed-pipe was opened and the contents pumped into the boiler (care being taken to empty one in less time than it required to fill the other) ; when empty, its feed-cock was closed. When the water in the tank reached within a few inches of the notch, a gong in the engine-room was sounded to call attention, and when it reached the notch the gong was struck twice ; at this instant the assistant engineer in the engine-room noted the reading of the counter, and an attendant in the fire-room noted and reported the height of water in the glass gauge on the boiler, as shown by a scale secured to it. The attendant at the tank also noted the time of filling, and the temperature when half emptied.

By this system of checks all errors of record could be detected, and it was possible to preserve and utilize any continuous run which came to an end through derangement of the engine. All parts of the tanks, pipes, and cocks were plainly visible to the eye, and had any leaks occurred therein they must have been detected.

A number of indicators were tested with steam before the trials, and a pair selected correct by a standard gauge. Indicator-diagrams were taken every twenty minutes, and the data for the columns of the log, except coal and ashes, every half hour.

It was ascertained that the pistons were tight by removing the cylinder-covers and letting on full steam-pressure.

During the first and principal experiments with each vessel the boilers were worked at their maximum power with natural draught at the dock, the fires cleaned regularly, and the cut-offs adjusted to carry a steam-pressure of about 70 pounds during

trial of the "Rush" and "Dexter," and about 35 pounds during that of the "Dallas."

At the conclusion of the principal experiments on each vessel, shorter experiments were made to determine the effect of varying the degree of expansion at the approximate steam-pressures of 70 and 40 pounds. In the case of the "Dexter" the cut-off was shortened for one experiment as much as the gear provided would permit; and for this vessel, as well as for the "Dallas," the cut-off was gradually lengthened, during other experiments, as far as the boiler would supply steam.

The long runs having demonstrated the evaporative qualities of the boilers, record was made, during the short runs, of the water used only. While these runs were in progress an officer was stationed at the tanks and one in the fire-room, in addition to the usual number on watch, to avoid the possibility of error.

The data obtained from these engines have been carefully classified and arranged by Prof. Cotterill.* They will be given presently.

Another good illustration of a marine-engine trial is the following, the report quoted being that of Sir F. J. Bramwell on the "Anthracite." As full an abstract is given as is needed to bring out the most salient parts and essential methods. In studying these results, it may be borne in mind that the efficiency of an ideal engine of similar working conditions, but free from wastes other than thermodynamic, would be about 0.26, and the weight of steam and fuel required, allowing an evaporation of 9 to 1, would be about 8 pounds and 0.9 pounds respectively.† The difference, 9 pounds of feed-water and 0.9 pounds of fuel, between the ideal and the real engine measures the wastes of the latter.

The engines are of the "direct-acting inverted" type, with surface condensation. Two cylinders are used; the after cylinder has two diameters of bore: the upper (the smaller one) is

* Cotterill's Steam-engine, p. 292 *et seq.*

† For details of computation see Rankine's Steam-engine, pp. 396-406, 410-411.

the high-pressure, and receives the steam from the boiler during the first half of the down stroke; the lower (the larger diameter) is the medium, or "intermediate," and is supplied at the up stroke with the steam which in the high-pressure did the work of the preceding down stroke. The exhaust from the bottom of the after cylinder passes into a chamber, from which is afforded the supply to the low-pressure (the forward) cylinder. Thus there is obtained in two cylinders an expansion of thirty-two times.

The surface condenser is composed of a number of close-topped, galvanized, wrought-iron tubes, standing vertically from a tube-plate, and having within them smaller tubes open at both ends, and proceeding upward from a lower tube-plate, so that the water from the sea passes up through the central tubes and down the annular spaces to the inlet of the circulating-pump.

The exhaust steam comes into contact with the exterior of the tubes, the condensed steam being drawn off by the air-pump, and returned to the hot-well, which surrounds the upper part of the condenser.

The space between the high-pressure piston and the upper side of the intermediate piston is in connection with the chamber which supplies the low-pressure cylinder.

The cylinders and their covers are heated by steam circulating through wrought-iron pipes cast into the thickness of the metal, and are very efficiently cleated, so as to prevent loss of heat.

The boiler is the Perkins boiler, formed of successive horizontal rows of wrought tubes (3 inches external diameter), connected at frequent intervals by vertical thimbles, the whole series being contained in a wrought-iron double casing, having the space filled in with vegetable black.

The boiler is supplied with distilled fresh water.

The drawings show the high-pressure cylinder to be 8 in. bore, the "intermediate" 16 in. bore, and the low-pressure 23 in. bore, but the cylinders were all somewhat smaller than the foregoing dimensions; the high-pressure cylinder is $7\frac{3}{4}$ in. diameter, the intermediate is $15\frac{1}{8}$ in., and the low-pressure is

22 $\frac{1}{8}$ in. The stroke in both cylinders is 1 ft. 3 in. The diameter of the piston-rods (the areas of which have to be deducted from the area of the intermediate piston, and from that of the underside of the low-pressure piston) is 2 $\frac{1}{4}$ in.

Preparations for the trial had been made by weighing out 50 cwt. of "Nixon's navigation hand-picked lumps," into 50 one-hundredweight sacks. These were ranged on deck. The bunkers, which were full, were sealed up both above and below. A spring-balance was hung up on deck, close to the stokehold hatch, and an assistant caused each sack of coal to be reweighed just before it was lowered for use. The sacks were afterwards separately weighed to obtain the net weight of the coal.

Fifty pounds of dry wood were served out, and two sacks of coal; and with these the fire was laid (the grate has an area of about 15 square feet).

The fire was lit..... at 6.28 A.M.

Steam was up and the engines were turned round at 7.18 A.M.

The height of water in the boiler-gauge was noted, and also the height in the hot-well; the still-cock being shut, was sealed in that position.

The steam stop-valve was sealed in its wide-open position.

The engines were started and the vessel got under weigh..... at 7.20 A.M.

The throttle-valve was put into the position which the engineer knew, from experience, would cause the engines to run at about 130 revolutions per minute after they became thoroughly heated up, and the handle was sealed into this position, the link motion being in full gear ahead.

The engines were provided with a Harding's counter, such as is used in the Navy, and there were pressure-gauges to show the boiler-pressure and the pressure in the chamber supplying the low-pressure cylinder, and the condenser was

provided with a vacuum-gauge. Four Richards indicators were fitted, viz.: one to the high-pressure end of the after cylinder, one to the intermediate end of that cylinder, one to the top and one to the bottom of the forward, the low-pressure, cylinder.

The first set of diagrams was taken..... at 8.22 A.M.

The first reading of the counter..... at 8.30 A.M.

From this time until..... 5.45 P.M.

the counter was read at the hours and half-hours, and sets of diagrams were taken at the quarter past the hour and at the quarter to the hour. The time when each sack of coals was lowered into the stokehold was noted, and also the time when the stoker commenced to use the contents of each sack.

The last shovelful of the 15th sack was put on... at 5.18 P.M.

and it was decided to stop the trial as soon as the coal then in the fire was exhausted.

The engines ran on until they stopped of themselves, and indicator-diagrams were taken, first at each quarter of an hour and then at each five minutes during the time they were gradually stopping.

The quarter-hour diagrams were taken until 6.30 P.M.

when the engines were making 124 revolutions, and the five-minute diagrams were taken until the engines came to a stand..... at 7.23 P.M.
or 12 hours 3 minutes after their start in the morning.

The water in the boiler was pumped to the same level in the gauge as that at which it had stood in the morning, and the height of water in the hot-well was noted.

The mean revolutions from 8.30 A.M. to 6.30 P.M., 10 hours, were 130.77 per minute, and from the first start to the same time being 11 hours 10

minutes, the mean revolutions were 130.4 per minute.

From the start at 7.20 A.M. to 6.30 P.M., 11 hours and 10 minutes, the engines developed an average gross indicated horse-power of 80.55, but from 6.30 to the time (7.23 P.M.) that the engines stopped of themselves, from the fire having burnt itself out, the power was gradually diminishing.

We estimate the 50 lbs. of wood as of about one-third the value of coal as fuel, say..... 17 lbs.
 The coal, 15 cwt..... 1680 "

 1697 lbs.

The loss of water for the whole 12 hours was $23\frac{1}{2}$ gallons.

The amount of lubrication was small, involving an expenditure of about one gallon of lard-oil, while cylinder and slide lubrications are in the Perkins system inadmissible and, with the metal used, unnecessary.

At the conclusion of the trial the assistants took away the four indicators, and the spring-balance with which the coals were weighed. All these were tested, with the result that the balance and the 100-lb. spring (used in the high-pressure cylinder-indicator) and some of the lighter springs were absolutely accurate, and that the variation in the others is too trifling to call for any allowance in calculating the mean pressures.

The mean pressures of the various diagrams were ascertained by dividing the areas of the diagrams obtained with the planimeter by the lengths of the diagrams.

The data obtained were as follows, as subsequently recomputed and arranged by the board appointed by the U. S. Navy Department to examine the vessel, the extraordinary pressures and ratios of expansion adopted attracting attention and making this work important :

Data and Results from the Experiment by Mr. Bramwell on the "Anthracite."

Economic results :

Pounds of coal consumed per hour per indicated
 horse-power..... 1.7114

| | |
|---------------------------------------------------------------------------|-------------|
| Pounds of coal consumed per hour per net horse-power..... | 1.9634 |
| Pounds of coal consumed per hour per total horse-power..... | 1.4291 |
| Pounds of combustible consumed per hour per indicated horse-power..... | 1.6259 |
| Pounds of combustible consumed per hour per net horse-power..... | 1.8653 |
| Pounds of combustible consumed per hour per total horse-power..... | 1.3577 |
| Pounds of feed-water consumed per hour per indicated horse-power..... | 17.8304 |
| Pounds of feed-water consumed per hour per net horse-power..... | 20.4560 |
| Pounds of feed-water consumed per hour per total horse-power..... | 14.8893 |
| Fahrenheit units of heat consumed per hour per indicated horse-power..... | 20,021.7027 |
| Fahrenheit units of heat consumed per hour per net horse-power..... | 22,969.9810 |
| Fahrenheit units of heat consumed per hour per total horse-power..... | 16,719.1503 |

Weight of steam accounted for by the indicator:

| | |
|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|----------|
| Pounds of steam present per hour in the first cylinder at the point of cutting off the steam, calculated from the pressure there..... | 989.3756 |
| Pounds of steam present per hour in the first cylinder at the end of the stroke of its piston, calculated from the pressure there..... | 890.6505 |
| Pounds of steam condensed per hour in the first cylinder to furnish the heat transmuted into the total horse-power developed in that cylinder by the expanded steam alone..... | 45.1639 |
| Sum of the two immediately preceding quantities | 935.8144 |
| Pounds of steam present per hour in the second | |

| | |
|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|------------|
| cylinder at the end of the stroke of its piston, calculated from the pressure there..... | 1,004.7334 |
| Pounds of steam condensed per hour in the first and second cylinders to furnish the heat transmuted into the total horse-power developed in those cylinders by the expanded steam alone..... | 124.8752 |
| Sum of the two immediately preceding quantities..... | 1,129.6086 |
| Pounds of steam present per hour in the third cylinder at the end of the stroke of its piston, calculated from the mean of the pressures there for the down-stroke and up-stroke of the piston..... | 1,118.3780 |
| Pounds of steam condensed per hour in the first, second, and third cylinders to furnish the heat transmuted into the total horse-power developed in those cylinders by the expanded steam alone..... | 199.1154 |
| Sum of the two immediately preceding quantities..... | 1,317.4934 |

Weight of water vaporized in the boiler from the feed temperature :

Pounds of steam evaporated per hour in the boiler on the supposition that this weight was equal to the weight of steam accounted for by the indicator at the end of the stroke of the piston of the third cylinder plus 121.9992 pounds condensed in that cylinder by other causes than the development of the power; this 121.9992 pounds is calculated from the weight of 147.2538 pounds condensed per hour in the third cylinder during the experiment made at the New York Navy-yard on the machinery of the "Anthracite," divided by the ratio 1.207, of the difference between

the temperatures of the initial steam in that cylinder on its piston, and of the back-pressure steam against it at the commencement of the stroke in that experiment and in the present one. In the navy-yard experiment the temperature of the initial steam on the piston of the third cylinder was 245.76 degrees Fahrenheit, and the temperature of the minimum back pressure against the piston was 150.25 degrees Fahrenheit; difference 95.51 degrees Fahrenheit. In Mr. Bramwell's experiment the temperature of steam of the initial pressure on the piston of the third cylinder was 230.60 degrees Fahrenheit, and the temperature of the minimum back pressure against it was 151.47 degrees Fahrenheit;

difference, $79^{\circ}.13$ Fahr. And $\frac{95.51}{79.13} = 1.207$,

the ratio used above..... 1,439.4926

Difference between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator :

Difference in pounds per hour between the weight of water vaporized (1455.9126 pounds) in the boiler and the weight of steam accounted for by the indicator in the first cylinder at the point of cutting off the steam.. 450.1170

Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the first cylinder at the point of cutting off the steam..... 31.27

Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the first cylinder at the end of the stroke of its piston..... 503.6782

| | |
|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|----------|
| Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the first cylinder at the end of the stroke of its piston..... | 34.99 |
| Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the second cylinder at the end of the stroke of its piston..... | 309.8840 |
| Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the second cylinder at the end of the stroke of its piston..... | 21.53 |
| Difference in pounds per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicator in the third cylinder at the end of the stroke of its piston..... | 121.9992 |
| Difference in per centum of the weight of water vaporized in the boiler between that weight and the weight of steam accounted for by the indicator in the third cylinder at the end of the stroke of its piston..... | 8.47 |

The results of the trials of the "City of Fall River," which follow, exhibit the effect of varying conditions of operation as a simple and as a compound engine.* This steamer was a side-wheel freight-boat of the Old Colony Steamboat Co., plying between New York, Newport, R. I., and Fall River, Mass., having a compound vertical-beam engine, H. P. cylinder 44"×8', L. P. cylinder 68"×12', built by W. & A. Fletcher, North River Iron Works, New York, 1883, and so constructed that the high-pressure cylinder could be entirely disconnected, leaving a simple beam engine, having a steam-cylinder 68"×12'.

* Report on Trial of the "City of Fall River," Jour. Franklin Inst.; July, 1881.

RESULTS OF EXPERIMENTS.

| Trip No. | Date. | Style of Engine. | Ports. | Distance, statute miles. | Draught of water. | Displacement, gross tons. | Running time. | Average pressure, steam per gauge. | Speed per hour. | Revolutions. | Revolutions per minute. | Horse-power. | Coal, in pounds. | Coal per hour, pounds. | Coal per hour per lbs. | Water per hour, in pounds. | Fed-temperature of water. | Water per hour per horse-power. | Tide, etc. |
|----------|----------|------------------|-------------------------|--------------------------|-------------------|---------------------------|---------------|------------------------------------|-----------------|--------------|-------------------------|--------------|------------------|------------------------|------------------------|----------------------------|---------------------------|---------------------------------|------------|
| | 1883. | | | | ft. in. | | h. m. | | Stat. miles. | | | | | | lbs. | | | lbs. | |
| 1 | May 3.. | Compound. | New York to Fall River. | 179 | 10 7 | 1,948 | 10 33 | 68½ | 16.96 | 15,833 | 25.01 | 1,578 | | | | 26,696 | 104.2 | 17 | Fair. |
| 2 | May 4.. | Compound. | Fall River to New York. | 179 | 10 5 | 1,908 | 11 35 | 69 | 15.45 | 17,711 | 25.5 | 1,615 | | 3,300 | 2.04 | 27,718 | 101.6 | 17.16 | Ahead. |
| 3 | May 9.. | Compound. | New York to Fall River. | 179 | 10 7 | 1,948 | 11 18 | 70 | 15.84 | 17,392 | 25.65 | 1,611 | 36,800 | 3,527 | 2.0217 | 27,834 | 97 | 17.3 | Ahead. |
| 4 | May 10. | Compound. | Fall River to New York. | 179 | 10 6½ | 1,938 | 10 46 | 70 | 16.62 | 16,669 | 25.8 | 1,622 | 35,609 | 3,307 | 2.0388 | 27,854 | 97 | 17.22 | Fair. |
| 5 | June 7.. | Simple..... | Fall River to New York. | 179 | 10 6 | 1,928 | 12 04 | 28 | 14.83 | 17,286 | 23.87 | 1,347 | | | .. | 35,271 | 111 | 26.2 | Ahead. |
| 6 | June 11. | Simple..... | New York to Newport... | 160 | 10 7 | 1,948 | 10 26 | 66 | 15.34 | 15,287 | 24.42 | 1,472 | 43,000 | 4,121 | 2.8 | | | | Even. |
| 7 | June 12. | Simple..... | Newport to New York. | 160 | 10 6 | 1,928 | 10 40 | 25½ | 15 | 15,415 | 24.1 | 1,457 | 44,150 | 4,139 | 2.84 | | | | Fair. |

In all cases the fires were well burnt down at the end of each trip, and, when the boat arrived at dock, fires were banked and coal put on to keep them alive while steam is blown off. During the day, about noon, more coal was put on the fires. An hour previous to departure fires were hauled down, and spread with fresh coal, to make steam, and ordinarily no further firing is necessary until half an hour after starting. On each trip indicator-cards were taken every half-hour. Water was measured by meters, readings taken every hour. The meters were tested by measuring water, under the same pressure as when feeding boilers, into a barrel, and weighing four cubic feet at a time; variations of such tests being from 61.4 to 61.5 lbs. per cubic feet. On June 10, the cut-off on the simple engine was shortened, making it easier to keep steam and to run with wide throttle. All water-measurements, power-calculations, and coal-measurements of trips Nos. 1, 2, 3, 4, 5 were reported to the Author by Messrs. Adger and Sague, the observers. The power and coal measurements of trips Nos. 6 and 7 were made by the W. & A. Fletcher Company.

Final Results.

Compound engine, 14 trips between New York and Fall River, May 15 to June 2. Average time, 11 hours $12\frac{6}{10}$ minutes; coal, 20.65 tons.

Simple engine, 12 trips between New York and Fall River, June 4 to 10. Average time, 11 hours $57\frac{6}{10}$ minutes; coal 27.42 tons.

Deducting 3 tons per trip for banking, spreading fires, donkey boiler, and kitchen (all of which is included in the amount of coal given), makes the actual consumption of coal per trip while the engine was running: for compound engine, 17.65 tons; and for simple engine, 24.42 tons.

The hull of this steamer was of the following dimensions:

| | |
|-----------------------------------------|---------|
| Length on the load water-line | 260 ft. |
| Length over all | 273 " |
| Breadth of beam on load water-line..... | 42 " |

| | |
|------------------------------------------------------------------------|-----------|
| Breadth of beam over guards..... | 73 ft. |
| Depth of hold moulded..... | 18 " |
| Draught of water, light..... | 9 " 3 in. |
| Draught of water, loaded 600 tons..... | 12 " |
| Depth between-deck, from top of plank-shear to top of upper frame..... | 11 " |

The paddle-wheels were of the feathering variety, and of the following proportions:

| | |
|-----------------------------------------------------------------------------------------|--------------|
| Diameter outside of buckets..... | 25 ft. 6 in. |
| Number of buckets..... | 12 |
| Width of each bucket..... | 40 in. |
| Length of each bucket..... | 10 ft. |
| Distance from centre of wheel-shaft to centre of eccentric actuating paddle-levers..... | 12 in. |
| Length of arm on bucket from axle to lever-pin..... | 21 in. |

The engine was found to have an efficiency of mechanism of about 83, and the paddles about 80 per cent.; their total efficiency being thus 66 per cent., or two-thirds. It was of the McNaught type. The boilers were tubular and contained—

| | |
|----------------------------------------------|----------------|
| Area of grate-surface in each boiler..... | 115 sq. ft. |
| Area of water-heating surface in each boiler | 3,345 " |
| Area of steam-heating surface in each boiler | 205 " |
| Ratio of water-heating to grate surface... | 29 |
| Ratio of total heating to grate surface... | 30.87 |
| Weight of each boiler..... | 51½ tons (net) |
| Weight of water in each boiler..... | 27 " " |

The figures in the table are deduced from their performance:

The thermodynamic efficiency of the steam as used in the compound engine on May 10 was computed according to the method given by Rankine.* The pressures of admission and of release and the mean back-pressure were obtained from the cards and the corresponding temperatures, densities, and latent heats arrived at by using the formulas given.†

* Steam-engine, § 284.

† *Ibid.*, §§ 206, 255.

RESULTS OF BOILER-TRIAL.

| Position of Boiler. | For'd. | Aft. | For'd. | Aft. | For'd. | Aft. |
|-----------------------------------------------------------------------------------------------|--------|--------|--------|--------|--------|--------|
| DATE OF TEST. | May 4 | May 4 | May 9 | May 9 | May 10 | May 10 |
| Length of test (hours)..... | 13 | 13 | 12.5 | 12.5 | 12.5 | 12.5 |
| Total coal burned (lbs.)..... | 18922 | 21314 | 18294 | 22083 | 18186 | 21650 |
| Total refuse, ash, etc. (lbs.)..... | 3699 | 3427 | 3518 | 3206 | 3449 | 3503 |
| Total combustible (lbs.)..... | 15223 | 17887 | 14776 | 18877 | 14737 | 18147 |
| Percentage of refuse, ash, etc..... | 19.55 | 16.08 | 19.23 | 14.52 | 18.97 | 16.18 |
| Total water evaporated (lbs.)..... | 154106 | 183785 | 151057 | 182019 | 144391 | 176576 |
| Average steam-gauge pressure..... | 68.5 | 68.5 | 70 | 70 | 70 | 70 |
| Average height of barometer..... | 30.70 | 30.70 | 30.70 | 30.70 | 30.5 | 30.5 |
| Average temperature of feed-water (Fhr.) | 102°.3 | 101° | 97°.1 | 97° | 67° | 97°.3 |
| Average temperature of atmosphere (Fhr.) | 74° | 74° | 78° | 78° | 77° | 77° |
| Average temperature of chimney gases (Fhr.)..... | 435° | 485° | 485° | 495° | 493° | 416° |
| Number of pounds of coal per hour per sq. ft. of grate..... | 12.657 | 14.257 | 12.726 | 15.292 | 12.646 | 15.069 |
| Number of pounds of water evaporated from the temperature of feed per lb. of coal..... | 8.14 | 8.62 | 8.257 | 8.24 | 7.939 | 8.156 |
| Number of pounds of water evaporated from the temperature of feed per lb. of combustible..... | 10.117 | 10.27 | 10.22 | 9.639 | 9.797 | 9.732 |
| Number of pounds of water evaporated from and at 212° F. per lb. of combustible..... | 11.59 | 11.77 | 11.75 | 11.08 | 11.266 | 11.192 |

The following are the data and results:

p_1 = absolute pressure of admission = 11808 lbs. per. sq. ft.

p_2 = absolute pressure of release = 1363.68 " "

p_3 = mean absolute back-pressure = 704.16 " "

t_1 = absolute temperature of feed-water = 558°.36 Fahr.

The corresponding temperatures, densities, and latent heats are designated by the same subscripts.

$t_1 = 774°.50$ Fahr.

$t_2 = 652°.32$

$L_1 = 131841.14$

$L_2 = 19000.39$

$D_1 = .1909$

$D_2 = .02606.$

From these data the following results were arrived at by considering the cylinders as non-conducting and the engine perfect:*

The ratio of expansion $r = 6.7167$.

Energy per cubic foot of steam admitted $UD_1 = 27183.43$ foot-lbs.

* Steam-engine, pp. 388, 389.

Heat expended per cubic foot of steam admitted $H_1 D_1$
 $= 163716.507$ foot-lbs.

Mean effective pressure, or energy per cubic foot swept through by piston,

$$\frac{UD_1}{r} = 4047.5 \text{ lbs. per sq. ft.}$$

Heat expended per cubic foot swept through by the piston,
 $\frac{H_1 D_1}{r} = 24377$ lbs. on square foot = pressure equivalent to heat expended.

$$\text{Efficiency of steam} = \frac{UD_1}{H_1 D_1} = \frac{U}{H_1} = .166.$$

Net feed-water per cubic foot swept through by piston

$$= \frac{D_1}{r} = .0284.$$

Cubic feet to be swept through by piston for each indicated horse-power per hour = $\frac{1980000}{M. E. P. = 4047.5} = 489.2$ cubic feet.

Feed-water per I. H. P. per hour = $489.2 \times .0284 = 13.89$ lbs.

Actual feed-water = 17.00 lbs., nearly.

13.89

Difference, 3.11 lbs. = 22 per cent.

due to cylinder condensation and leakage waste and other wastes not taken into account. This is an exceptionally excellent result. These wastes are usually much greater.

101. Results and Deductions and Graphical Records of these trials are rich in matters of interest to the engineer. The data obtained in the trials of the engines designed by Mr. Emery are as follows: *

* Cotterill, pp. 294-296.

DISTRIBUTION OF HEAT.

| DESCRIPTION OF ENGINE. | PRINCIPAL PARTICULARS. | | | USEFUL HEAT EXPENDED PER CENT. OF TOTAL. | | | LOSSES OF HEAT PER CENT. OF TOTAL HEAT. | | | | TOTAL EXPENDITURE OF HEAT AND STEAM. | | |
|--------------------------------------------------------|------------------------|------------------|---------------------|------------------------------------------|---------------------------|--------------|-----------------------------------------|-----------------------|----------------|-----------------------|--------------------------------------|-------------------------------|------------------------------------|
| | Boiler Pressure. | Speed of Piston. | Ratio of Expansion. | Useful Work. | Loss in a Perfect Engine. | Useful Heat. | Exhaust Waste. | Incomplete Expansion. | Heat-ing Feed. | Excess Back-pressure. | Other Losses. | Lbs. Steam per I. H. P. Hour. | Thermal Units per I. H. P. per 1'. |
| "BACH." | | | | | | | | | | | | | |
| 1. Compound engine, low-pressure cylinder, jacketed. | 95½ | 150 | 11½ | 6.7 | 20.6 | 20.3 | | | | | | 35.1 | 633 |
| 2. Operated as a simple engine without jacket. | 94½ | 180 | 7½ | 8.0 | 24.4 | 32.4 | 37.6 | 11.2 | 7.9 | 2.6 | 8.3 | 29.6 | 534 |
| 3. Operated as a simple engine without jacket. | 92½ | 188 | 5½ | 8.9 | 27.1 | 36.0 | 27.2 | 18.0 | 7.9 | 2.4 | 8.5 | 26½ | 480 |
| 4. Operated as a simple engine, with jacket in use. | 95½ | 160 | 12½ | 8.6 | 24.4 | 33.0 | 32.0 | 12.8 | 8.0 | 1.8 | 12.4 | 27.1 | 493 |
| 5. Operated as a simple engine, with jacket in use. | 95½ | 185 | 8½ | 9.7 | 27.5 | 37.2 | 27.9 | 13.4 | 8.1 | 1.5 | 11.9 | 24.1 | 441 |
| 6. Operated as a simple engine, with jacket in use. | 94 | 215 | 5 | 10.2 | 29.2 | 39.4 | 21.9 | 21.2 | 8.2 | 2.1 | 7.2 | 23.1 | 419 |
| 7. Operated as a simple engine, with jacket in use. | 45½ | 181 | 2½ | 7.1 | 30.6 | 37.7 | 24.4 | 22.9 | 6.4 | 3.3 | 5.3 | 34.0 | 602 |
| 8. Operated as a compound engine, with jacket in use. | 96½ | 155 | 17 | 9.2 | 24.3 | 33.5 | 15.4 | 15.6 | 7.6 | 4.0 | 23.9 | 25.1 | 465 |
| 9. Operated as a compound engine, with jacket in use. | 95½ | 193 | 9½ | 11.0 | 28.2 | 39.2 | 13.5 | 18.5 | 8.0 | 2.5 | 18.3 | 20.7 | 389 |
| 10. Operated as a compound engine, with jacket in use. | 95 | 213 | 7 | 11.3 | 28.8 | 40.1 | 16.4 | 19.1 | 8.2 | 2.5 | 13.7 | 20.3 | 378 |
| 11. Operated as a compound engine, with jacket in use. | 95 | 225 | 5½ | 11.2 | 28.4 | 39.6 | 15.6 | 19.6 | 8.2 | 2.5 | 14.5 | 20.4 | 382 |
| 12. Operated as a compound engine, with jacket in use. | 94 | 242 | 4½ | 11.1 | 30.8 | 41.9 | 15.0 | 19.9 | 8.0 | 1.9 | 13.3 | 21.2 | 385 |
| "DALLAS." | | | | | | | | | | | | | |
| 13. Simple engine, not jacketed. | 50 | 243 | 5 | 9.4 | 33.1 | 42.5 | 20.4 | 18.1 | 6.8 | 3.2 | 9.0 | 26.7 | 455 |
| 14. Simple engine, not jacketed. | 50 | 285 | 3½ | 9.4 | 33.1 | 42.5 | 16.3 | 20.8 | 6.8 | 3.6 | 10.0 | 27.0 | 455 |
| 15. Simple engine, not jacketed. | 46½ | 308 | 3½ | 9.4 | 35.8 | 45.2 | 17.8 | 22.1 | 6.2 | 4.3 | 4.4 | 26.9 | 455 |
| 16. Simple engine, not jacketed. | 48½ | 322 | 2½ | 8.9 | 33.5 | 42.4 | 19.6 | 21.9 | 6.7 | 3.8 | 5.7 | 28.9 | 480 |
| 17. Simple engine, not jacketed. | 42 | 317 | 2½ | 8.3 | 33.0 | 41.3 | 19.4 | 23.9 | 6.4 | 4.0 | 5.0 | 31.0 | 515 |

DISTRIBUTION OF HEAT.—Continued.

| DESCRIPTION OF ENGINE. | PRINCIPAL PARTICULARS. | | | USEFUL HEAT EXPENDED PER CENT. OF TOTAL. | | | LOSSES OF HEAT PER CENT. OF TOTAL HEAT. | | | | | TOTAL EXPENDITURE OF HEAT AND STEAM. | |
|------------------------|------------------------|------------------|---------------------|------------------------------------------|---------------------------|--------------|-----------------------------------------|-----------------------|---------------|-----------------------|---------------|--------------------------------------|------------------------------------|
| | Boiler Pressure. | Speed of Piston. | Ratio of Expansion. | Useful Work. | Loss in a Perfect Engine. | Useful Heat. | Exhaust Waste. | Incomplete Expansion. | Heating Feed. | Excess Back-pressure. | Other Losses. | Lbs. Steam per I. H. P. per Hour. | Thermal Units per I. H. P. per i'. |
| | | | | | | | | | | | | | |
| "DEXTER." | 83½ | 339 | 4½ | 10.4 | 29.7 | 40.1 | 21.0 | 19.1 | 8.2 | 3.9 | 7.7 | 22.6 | 411 |
| | 82 | 366 | 3½ | 10.4 | 30.0 | 40.4 | 20.0 | 21.1 | 8.2 | 3.9 | 6.4 | 22.7 | 411 |
| | 81½ | 437 | 2¾ | 10.3 | 30.6 | 40.9 | 13.7 | 25.2 | 8.0 | 5.2 | 7.0 | 23.0 | 415 |
| | 55½ | 305 | 3½ | 8.6 | 27.1 | 35.7 | 24.9 | 20.2 | 7.4 | 4.5 | 7.3 | 27.4 | 497 |
| | 56½ | 364 | 2 | 7.8 | 26.4 | 34.2 | 26.4 | 21.7 | 7.3 | 4.4 | 6.0 | 30.2 | 548 |
| "RUSH." | 84 | 319 | 6½ | 12.7 | 36.3 | 49.0 | | | | | | 18.4 | 336 |
| | 51½ | 250 | 4 | 10.6 | 36.3 | 46.9 | | | | | | 22.1 | 404 |
| "GALLATIN." | 86½ | 255 | 7.3 | 11.4 | 32.3 | 43.7 | 15.2 | 20.1 | 7.8 | 3.1 | 10.1 | 20.5 | 374 |
| | 82 | 343 | 4.9 | 10.9 | 32.9 | 43.8 | 12.9 | 25.4 | 7.8 | 3.1 | 7.0 | 21.5 | 390 |
| | 60½ | 221 | 6.1 | 10.2 | 30.1 | 40.3 | 13.9 | 22.7 | 7.2 | 5.0 | 10.9 | 22.9 | 420 |
| | 230 | 5.1 | 9.8 | 30.1 | 40.9 | 20.3 | 17.8 | 17.8 | 6.7 | 3.7 | 10.6 | 24.0 | 437 |
| | 56 | 246 | 3.7 | 10.2 | 33.8 | 44.0 | 18.3 | 25.7 | 7.1 | 3.9 | 1.0 | 23.2 | 420 |
| | 52½ | 291 | 2.2 | 9.0 | 32.8 | 41.8 | 13.8 | 27.1 | 6.7 | 3.5 | 7.1 | 26.5 | 476 |
| | 30 | 206 | 2.0 | 7.2 | 31.9 | 39.1 | 16.9 | 26.6 | 5.8 | 4.3 | 7.3 | 33.3 | 591 |
| | 28 | 212 | 1.5 | 6.4 | 30.4 | 36.8 | 17.7 | 25.6 | 5.4 | 4.7 | 9.8 | 37.4 | 604 |
| | 86 | 261 | 7.8 | 9.4 | 26.6 | 36.0 | 22.8 | 19.5 | 8.5 | 3.4 | 9.8 | 25.0 | 454 |
| | 83½ | 300 | 5.0 | 10.6 | 29.7 | 40.3 | 17.8 | 24.4 | 8.3 | 2.0 | 7.2 | 21.9 | 401 |
| | 59½ | 215 | 5.9 | 9.0 | 26.9 | 35.9 | 21.5 | 14.7 | 7.7 | 3.0 | 14.2 | 26.0 | 476 |

DISTRIBUTION OF HEAT.—Continued.

| DESCRIPTION OF ENGINE. | PRINCIPAL PARTICULARS. | | | USEFUL HEAT EXPENDED PER CENT. OF TOTAL. | | | LOSSES OF HEAT PER CENT. OF TOTAL HEAT. | | | | | TOTAL EXPENDITURE OF HEAT AND STEAM. | |
|----------------------------------------------------------------------|------------------------|------------------|---------------------|------------------------------------------|---------------------------|--------------|-----------------------------------------|-----------------------|---------------|-----------------------|---------------|--------------------------------------|------------------------------------|
| | Boiler Pressure. | Speed of Piston. | Ratio of Expansion. | Useful Work. | Loss in a perfect Engine. | Useful Heat. | Exhaust Waste. | Incomplete Expansion. | Heating Feed. | Excess Back-pressure. | Other Losses. | Lbs. Steam per I. H. P. per Hour. | Thermal Units per I. H. P. per 1'. |
| "GALLATIN."—(Cont'd.) | 36. { 58 | 254 | 3.7 | 9.8 | 30.3 | 40.1 | 17.7 | 24.9 | 7.5 | 2.0 | 7.8 | 24.0 | 437 |
| | 37. { 52½ | 278 | 2.2 | 8.6 | 35.4 | 44.0 | 16.0 | 22.0 | 6.3 | 2.0 | 9.7 | 28.1 | 596 |
| | 38. { 29½ | 200 | 2.0 | 5.9 | 25.8 | 31.7 | 31.5 | 21.9 | 5.9 | 4.1 | 4.9 | 40.4 | 720 |
| | 39. { 27½ | 204 | 1.5 | 5.4 | 25.8 | 31.2 | 28.7 | 24.8 | 5.4 | 4.0 | 5.9 | 44.2 | 787 |
| Jacket supplied with steam of 85 lbs. Difference of temperature 68°. | 40. { 28½ | 205 | 1.8 | 6.9 | 27.5 | 34.4 | 15.0 | 31.9 | 5.8 | 4.1 | 8.8 | 34.1 | 616 |
| | 41. { 29 | 211 | 1.5 | 7.1 | 29.5 | 36.6 | 13.6 | 33.7 | 5.7 | 4.4 | 6.0 | 34.9 | 605 |
| Without vacuum; jacket in use. | 42. { 84½ | 247 | 4.1 | 9.9 | 64.1 | 74.0 | 4.0 | 1.8 | 4.7 | | 15.5 | 25.9 | 431 |
| | 43. { 82 | 266 | 3.5 | 9.4 | 61.7 | 71.1 | 6.7 | 2.7 | 4.5 | | 15.0 | 27.3 | 456 |
| Without vacuum; jacket not in use. | 44. { 85½ | 233 | 4.4 | 8.6 | 55.4 | 64.0 | 13.4 | 1.5 | 4.9 | | 16.2 | 30.0 | 499 |
| | 45. { 81 | 258 | 3.5 | 8.8 | 58.1 | 66.9 | 10.1 | 3.8 | 4.5 | | 14.7 | 29.4 | 488 |

The expenditure of heat in the last column is reckoned from the temperature of the feed-water. The number obtained by dividing 42.75 by the expenditure of heat thus reckoned is the true efficiency of the engine. Experiments 42-45 are exceptions, made without vacuum. In calculating the expenditure of heat, the boiler has been supposed to supply dry steam; the results obtained are too large, though not much too large, as there is no reason to believe the amount of priming considerable.

The columns headed "Useful Heat Expended" show the useful work done, together with the corresponding *necessary* loss, expressed as a percentage of the total heat expended. The first of these columns is the absolute efficiency of the engine, and the third the efficiency relatively to a *perfect* engine working between the same limits of temperature.

The five remaining columns show heat unnecessarily lost—

(1) By the exhaust waste, by transmission of heat to the exhaust steam, and by external radiation less the heat given out by piston friction and the effects of compression.

(2) By incomplete expansion, by the amount of work which the steam discharged from the cylinder might do by expanding down to the pressure of the condenser.

(3) By misapplication of heat in heating the feed, raising the temperature of the water by direct heat instead of by compression of the exhaust steam.

(4) By excess back-pressure, the difference between the actual back-pressure and the pressure corresponding to the temperature of the condenser.

(5) By other losses, by clearance and wire-drawing and by misapplication of heat during expansion.

All the results are given as percentages of the total expenditure; but by multiplying by that expenditure, and dividing by 100, they may be expressed in thermal units per I. H. P. per minute; or, by multiplying by the consumption of steam and dividing by 100, they may be expressed in pounds of steam per I. H. P. per hour.

These computations have been made very carefully by Pro-

fessor Cotterill, and interesting conclusions are reached by their study: The large losses by exhaust waste, 27 per cent., when in the first set of these trials the steam-jacket was dispensed with, and the exaggeration of that loss by increased ratios of expansion; the reduction of this loss in the next set of four trials, by the use of the jacket; the increase of waste invariably, with increase of surface exposed; the great gain in this direction by compounding, as seen in the final set of five "Bache" trials; while the reduction of exhaust waste is partially compensated by increased liquefaction in the high-pressure cylinder. The net final advantage of compounding all these effects and variations of condition are well shown by these data. The work may be studied in detail in the treatise from which the figures are quoted. The trials of the "Dexter" developed the gain by increased piston-speed which might have been anticipated; and those of the "Rush" exhibited, again, the gain due to compounding.

The cylinders of the compound engine of the "Rush" were steam-jacketed, and those of the non-compound engines were not steam-jacketed. A new non-compound engine with steam-jacketed cylinder and a boiler designed for high-pressure steam having been completed for the revenue-steamer "Gallatin," a series of trials of the machinery of that vessel, with and without steam-jacket in use, for comparison with the trials detailed in the report, were made, and the data are similarly arranged for comparison in Nos. 25 to 45 of the table. The first twelve or fifteen trials illustrate again the good effect of steam-jacketing in reducing wastes and permitting a great increase in the best ratios of expansion at any one pressure in the compound engine. The "Gallatin" trials show similar effects in the non-compound engine. This latter engine consisted of a pair of single cylinders 34 inches in diameter and 30 inches stroke; and the trials were conducted substantially as were those of the other ships. Examining the table, it is seen that all the earlier of this set of trials exhibit in a very decided manner the good effect of the jackets, reducing waste by cylinder condensation, and increasing the ratios of expansion at maximum efficiency,

